AD-751 217

HIGH PERFORMANCE HELICOPTER HOIST PROGRAM

Louis J. Lawson, et al

Lockheed Missiles and Space Company

Prepared for:

Army Land Warfare Laboratory

August 1972

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Lockheed Missiles & Space Company
Sunnyvale, California

August 1972

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U.S. ARMY LAND WARFARE LABORATORY
Aberdeen Proving Ground, Maryland

UNCLASSIFIED
Security Classification

DOCUMENT CONTROL DATA - R & D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY (Corporate author) Lockheed Missiles & Space Company, Inc. 1111 Lockheed Way Sunnyvale, California 94088	2a. REPORT SECURITY CLASSIFICATION Unclassified 2b. GROUP N/A

3. REPORT TITLE

HIGH PERFORMANCE HELICOPTER HOIST PROGRAM
(Phase I – Final Report)

4. DESCRIPTIVE NOTES (Type of report and inclusive dates)

5. AUTHOR(S) (First name, middle initial, last name)

Louis J. Lawson

6. REPORT DATE	7a. TOTAL NO. OF PAGES	7b. NO. OF REFS
15 May 1972	152	14

8a. CONTRACT OR GRANT NO. DAAD05-72-C-0099	9a. ORIGINATOR'S REPORT NUMBER(S) LMSC-D267477
b. PROJECT NO.	
c.	9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report)
d.	

10. DISTRIBUTION STATEMENT

Each transmittal of this document outside the agencies of the U. S. Government must have prior approval of the U. S. Army Land Warfare Laboratory.

11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY
	U. S. Army Land Warfare Laboratory Aberdeen Proving Ground Maryland 21005

13. ABSTRACT

This report presents a feasibility study on a flywheel powered personnel rescue hoist for helicopters. It includes investigations of flywheel configurations, materials, and associated equipment including bearings, seals, and pumps. Also included are discussions of human factors considerations and design considerations affecting component selection, hoist configuration. and hoist installation. Particular attention is given to the capability of a 13.5-lb, 73.5 w-hr flywheel to provide the short duration high power required to affect rescue hoisting speed as high as 500 ft per min. without overtaxing the helicopter's limited accessory power capacity.

Details of illustrations in this document may be better studied on microfiche

DD FORM 1 NOV 65 1473

UNCLASSIFIED
Security Classification

UNCLASSIFIED
Security Classification

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Hoist						
Flywheel Energy Storage						

ib

UNCLASSIFIED
Security Classification

TECHNICAL REPORT NO. LWL-CR-02M72

HIGH PERFORMANCE HELICOPTER HOIST PROGRAM

PHASE I

FINAL REPORT

Contract No. DAAD05-72-C-0099

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August 1972

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SUMMARY

The purpose of the Phase I, High Performance Helicopter Hoist Program is to determine the feasibility of utilizing flywheel energy storage to provide high speed retrieval of personnel and materiel while hovering. A further objective is the provision of a preliminary design of the optimum hoist configuration which will interface with existing and new U. S. Army helicopters used for air rescue operations.

RESULTS AND CONCLUSIONS

An optimized hoist design layout has been completed which will operate at hoisting speeds up to five times faster than the present hoist without additional power from the helicopter during hoisting operations.

A moderate capacity kinetic energy flywheel weighing 13.5 lb can provide sufficient power during hoisting operations to minimize helicopter vulnerability.

A practical transmission system coupling the drive, motor, flywheel, and cable mechanism can be built.

Tests of candidate steel cables and ropes for use with the hoist have shown synthetic rope to offer several advantages.

Human factors studies and testing have pointed up many deficiencies in the presently used hoisting methods and equipment which can be improved with a new hoist system.

The life-cycle cost and cost effectiveness of the optimized high performance hoist are predicted to be substantially below present levels as a result of improved availability and reduced flight time for rescue operations.

A safety analysis has shown that the high performance hoist can alleviate hazards associated with hoist rescue operations without introducing significant new hazards.

RECOMMENDATIONS

A two-phase program leading to the design and fabrication of a high performance helicopter hoist and ancillary equipment suitable for military potential testing is recommended.

The recommended prototype hoist development program includes the following:

- Human factors analysis of entire hoisting system
- Verification testing of rope handling characteristics
- Design and fabrication of high performance hoist
- Design and fabrication of improved traction sheave for rope lowering
- Comprehensive testing of hoist system prior to military potential tests

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Section 1
INTRODUCTION

NEED FOR IMPROVED HOIST

The ascent and descent speed of existing helicopter hoists employed by the U. S. Army in medical evacuation and rescue service is limited to approximately 100 fpm. During hover, the helicopter, its personnel, and the rescuee are highly vulnerable to enemy attack. This vulnerability can be reduced by increasing the operating speed of the hoist so as to minimize hover time. The existing limitation in hoist speed is imposed basically by constraints on available power from the helicopter and by the need for keeping the weight of the hoist to a minimum. The conflicting requirements for holding power consumption and system weight to present levels while substantially increasing hoist speed can only be met by the incorporation of an energy storage system into the hoist. Thus, energy could be taken from the aircraft electrical system enroute to a rescue mission and stored for use in effecting high power retrieval operations.

The constraints imposed on the weight of the present hoist have necessitated operation of the electric drive motor into its overload rating even for 100 fpm hoisting operations. Thus, only a limited duty cycle is available prior to the need to wait for the motor to cool. This limitation can result in the need for added hovering or circling time to a rescue mission.

With the hoist powered by stored energy, the full capabilities of the aircraft's electrical and hydraulic systems are available for electronic, control boost, illumination, and other equipment which may be required to function during medical evacuation and rescue operations.

With a substantially increased hoisting speed, less hover time is required to pick up a given number of men from a given height. This, in turn, reduces: (a) power taken from the engine, (b) the probability of an engine failure during hover, (c) mission time, (d) fuel consumption, and (e) attrition due to enemy action. Conversely, more men can be evacuated from a given spot before hostile action is brought against the hovering or loitering aircraft. High speed hoists can make rescue operations from higher hover heights feasible where this is desired.

During rescue or evacuation hoist operations (whether in a combat zone or not), the helicopter is almost always in an undesirable operating region, as defined by the so-called "dead man's curve." To illustrate this point, the Height-Velocity Diagram for a UH-ID aircraft is shown in Fig. 1-1. It may be seen that except for hovering operations in a very high wind, the aircraft is in an unsafe operating zone during rescue operations where an engine malfunction would result in loss of aircraft and crew.

KINETIC ENERGY AS A SOLUTION

In the search for a practical energy storage means with near-term availability which would be suitable for the helicopter hoist application, consideration was given to several techniques. Most of these applicable techniques have been previously applied to vehicle propulsion. Recent studies (1 and 2)* and hardware development programs (3 and 4) have resulted in characteristics data based on presently available energy storage systems in suitable form for comparative evaluation. A summary tabulation of these data is shown in Table 1-1.

The excellent match of kinetic energy storage characteristics to the high performance hoist demands became obvious on the basis of the comparative data with other storage techniques. The flywheel can provide highly accept-

*Numbers in parentheses designate references which are listed in Section 10 of this report.

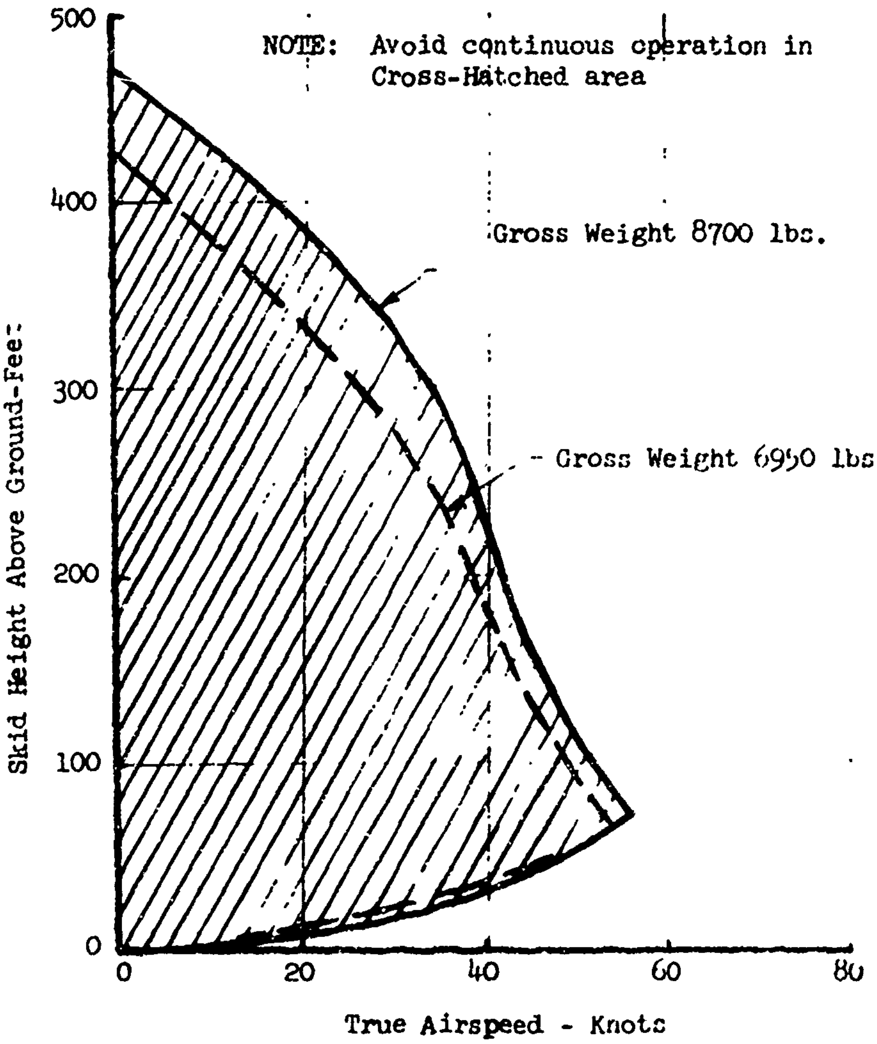


Fig. 1-1 Height-Velocity Diagram for UH-1D
(Density Altitude = 2, 300 ft)

Figure 1-1

Table 1-1
ENERGY STORAGE COMPARISON

Storage Phenomena	Energy Density (whr/lb)	Power Density (w/lb)	Deep (75%) Discharge Cycle Life (Cycles)
Compression of Gases	1.51	>5000	$>10^7$
Hydraulic Accumulator	3.5	>5000	$>10^7$
Elastic Deformation:			
Steel Spring	0.04	>1000	$>10^7$
Natural Rubber Band	4.00	35	1000-5000
Electrochemical Reaction:			
Lead-Acid Battery	8	35	300-500
Nickel-Cadmium Battery	14	35	1000-3000
Kinetic Energy:			
Maraging Steel Flywheel	25	>5000	$>10^7$
AISI 4340 Steel Flywheel	15	>5000	$>10^7$

able energy density levels which favorably influence hoist weight or operating duty cycle. At the same time, the flywheel can provide high levels of power to make available the high power required for high speed retrievals which is desirable in helicopter rescue operations. The high flywheel energy density makes possible rapid recharging or spin-up. In addition, the flywheel has almost unlimited deep discharge cycle life which is fully compatible with the 10-year design service life requirements of military equipment. The review of comparative data presented in Table 1-1 revealed the flywheel to be the only practical energy storage means with the desired combination of characteristics.

The effectivity provided by the use of flywheel energy storage in the hoist system offers several operational advantages for medical evacuations and rescue operations. For example, the energy stored in the flywheel can be discharged rapidly to effect high cable speeds, and can be charged as slowly

as desired. This being the case, the storing of energy can be accomplished by using a smaller portion of the electric, hydraulic, or mechanical power available within the aircraft. Furthermore, the flywheel system can continue to operate in case of failure of an aircraft's electrical or hydraulic systems.

PURPOSE OF DESIGN STUDY AND TRADEOFFS PROGRAM

The first phase of the High Performance Helicopter Hoist Program which is described in this report is intended to provide design study information and technical tradeoffs data suitable to determine the feasibility of applying kinetic energy storage to an improved helicopter hoist capable of being used by the U. S. Army aboard helicopters presently in the field as well as anticipated new helicopter systems. The major thrust of this program is directed toward improvement of medical evacuation operations in combat zones although attention is given to supply and rescue missions involving lowering and lifting of personnel and strategic materiel.

The specific objectives of the design study and tradeoffs program are the following:

- Assessment of the operational deficiencies of the present U. S. Army helicopter hoist, medical evacuation methods, and rescue ancillaries
- Determination of the helicopter/hoist interfaces and constraints especially for the UH-1H helicopter as currently used for medical evacuation missions
- Assistance to the U. S. Army Land Warfare Laboratory (USALWL) in establishing realistic military characteristics for a high speed utility helicopter hoist
- Review of flywheel technology to determine the optimum capacity and configuration of the flywheel assembly for the hoist

- **Determination of the most feasible drive configuration for the flywheel and cable handling systems**
- **Preparation of preliminary layouts of the optimum high performance hoist assembly**
- **Recommendation of subsequent program phases (if feasibility is shown) leading to a prototype hoist suitable for military potential testing.**

Section 2
APPROACH

The High Performance Helicopter Hoist Phase I Program conducted for the USALWL was intended to be an objective assessment of the feasibility of an improved utility hoist using kinetic energy storage. The technical approach followed by Lockheed Missiles & Space Company, Inc. (LMSC) in conducting this program is described in the succeeding paragraphs.

PRESENT EQUIPMENT AND OPERATING PROCEDURES

At the beginning of the program an examination was made of the present rescue hoist and its operation. This investigation included the helicopter, boom, traction sheave, pendant, swivel, intercom system, hook, jungle penetrator, etc., as well as the hoist itself. To gather data on the present system trips were made to U. S. Army Combat Developments Command (USACDC), Fort Belvoir, Virginia; USACDC Medical Service Agency, Fort Sam Houston, Texas; Crissy Field, Sixth Army Headquarters, San Francisco Presidio (three trips); and Bell Helicopter Company, Fort Worth, Texas. An additional trip was made to Western Gear Company, Lynwood, California, to inspect a next-generation hoist which is under development. A Breeze hoist, used in the present system, was requisitioned by USALWL for LMSC use and tests were conducted. A mockup of the UH-1H cabin area was made and the hoist installed for space, time, and motion evaluations. From this review of the present system it was concluded that the program should be widened in scope. The modified approach used in conducting the Phase I program is shown in the flow diagram of Fig. 2-1. The present hoist system review made it apparent that present equipment falls short of meeting even current requirements in terms of performance and reliability. From this it was concluded that the techniques and materials presently employed could not be projected to achieve the performance

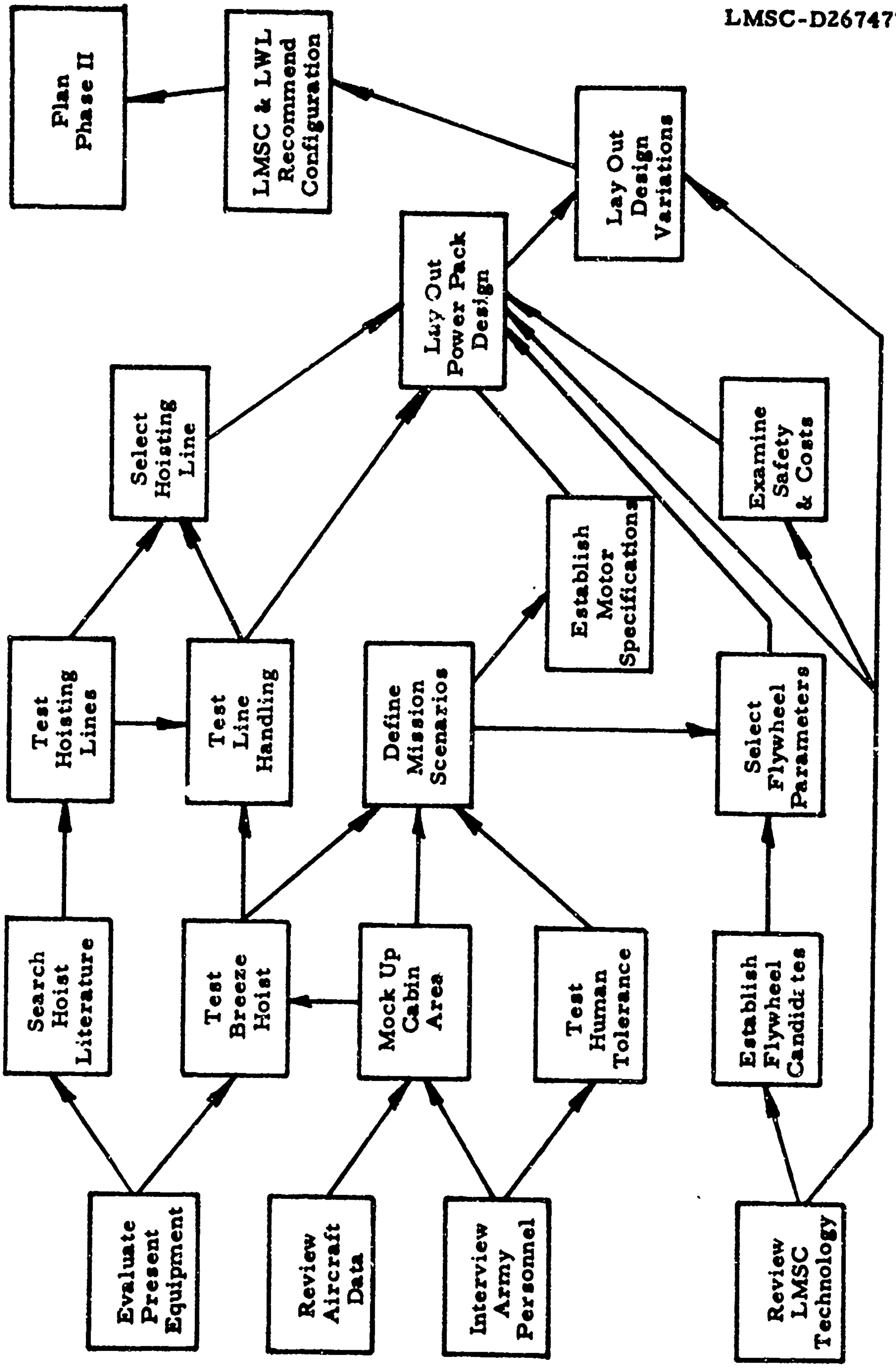


Fig. 2-1 Approach

desired in a next generation rescue hoist. Not only should new technology be applied to the power supply, but also to the hoisting line and line handling. Although not included in the scope of the present study, a well-balanced, high performance hoist system should eventually include improvements in the man-machine interfaces such as evacuee attachment and operator controls.

The review of present equipment and its operation also formed a basis for judgements as to the degree of improvement which could be reasonably expected of the next generation.

POWER COMPONENT PARAMETERS

Based on the desired improvement in the state of the art, typical mission scenarios were defined by LMSC in conjunction with USALWL. Testing with human subjects verified the feasibility of increased hoisting rates which interviews with Army personnel with a variety of experience indicated to be desirable. The number of lifts per hover, the loads, and the height of the lift established the mission hoist energy requirement.

Energy for the high performance hoist is supplied by the helicopter turbine engine-driven generator which powers the electric hoist motor. The motor, in turn, drives the flywheel which stores energy to provide for peak power demands. In this manner the performance drawbacks of the present hoist motor are overcome. The shunt-type motor chosen for this drive has a fixed maximum speed making runaway impossible. Inasmuch as the motor is not required to operate in an overload condition during heavy, high lifts, it is no longer necessary to wait to cool between hoist cycles. Estimates of hoist dwell times between lifts which assumed continued use of present evacuee attachment equipment and techniques were made in proportioning the motor and the flywheel. Based on the scenarios the output of the motor was established as well as the storage capacity of the flywheel, and a motor specification was written.

A computer-aided analysis presented a spectrum of flywheel candidates, which were of a modified hyperbolic shape, selected for its minimum weight characteristic. A number of these candidates were eliminated by a maximum diameter constraint imposed by the requirement that the bulkiness of present equipment not be exceeded. Selection of the flywheel was made based on a two-to-one usable speed ratio, which was judged to retain a reasonable reserve. A review was made to assure that the design speed was well within seal and bearing limits. The flywheel material was selected to provide the low stress level required by a safety analysis.

HOISTING LINE INVESTIGATION

In broadening the scope of the High Performance Hoist Program, investigation was made into hoisting line materials and hoisting line handling techniques. This investigation was in the form of literature search, vendor contact, and shop testing. In the test evaluation special attention was given to the capstan drive (at both present and desired speeds) and to 3/16 in., 19 x 7 nonrotating, stainless steel wire rope, since they were bases for comparisons. In addition to other types of wire rope, various synthetic ropes and webbings were included. Webbing had the distinct attraction of avoiding the level winding problem. Tests and other types of evaluation of hoisting materials included an examination of the following characteristics:

- Elongation
- Shock attenuation
- Traction
- Cold flow
- Speed capability
- Bending fatigue
- Abrasion resistance
- Frictional heating
- Fleeting
- Stretch induced spin
- Rotational damping
- Bounce damping
- Torsional stiffness
- Whipping
- Fleeting characteristics
- Spooling characteristics
- Kinking
- Extreme temperature effects
- Cut vulnerability

DESIGN LAYOUT

A preliminary design layout was made using the motor specification and the characteristics selected for the flywheel. The flywheel housing, ancillary equipment, gear reductions, brake, clutch, and storage drum were included. The storage drum inner and outer diameters (and hence the total reduction ratio) were both dependent on the hoist line material selection. Design decisions were made to use a controllable dissipative clutch and spur gearing reduction. Layout variations were established, using this basic "power pack" assembly, to indicate aircraft installation options, such as pylon and separate unit mountings and hoist line handling and material options. USALWL and LMSC joined in selecting from these variations the most promising combination recommended for design, fabrication, and testing of a prototype hoist in a subsequent program phase.

Section 3
REQUIREMENTS STUDY

The principal areas of study necessary to define the operational requirements of the high speed helicopter rescue hoist are described in the following paragraphs.

HOIST CONTROL REQUIREMENTS

The rescue hoist is a part of a large, complex, integrated man-machine system consisting of: the crew aboard the aircraft; the helicopter; ground personnel and equipment at the aircraft's home base; communication equipment; a variety of devices for attaching the evacuee; and, ground personnel at the rescue site.

The pilot has communication links with his copilot, his home base, the hoist operator, and ground personnel at the rescue site. He also controls a guillotine switch for cutting the hoisting line in case of an emergency.

The hoist operator is in communication with both the cockpit and the ground. In addition to the guillotine switch, he also controls: a 3-way, variable switch for ascent, stop, and descent; a 2-way switch for extending and retracting the hoist boom, and a hoist on-off switch over and above these operator actuated controls, the machine itself performs various automatic control functions. The hoist units, which are eventually controlled, are the drive motor, boom actuator, clutch, brake, descent drive, and guillotine.

The following typical hoist operations indicate the interrelationship of manual and automatic controls.

Descent Command

With the storage drum stopped, the flywheel declutched, and the hoist line either fully or partly reeled in, the operator moves the control switch in the "descent" direction releasing the brake and gradually increasing the payout speed by actuating a variable speed drive (separate from the ascent drive). Descent speed, from creep to maximum, is regulated by the control unit utilizing a tachometer which senses line speed and provides feedback data to the control loop. This provides for variations in load from zero to maximum weight.

Ground Proximity Problem

Consideration was given to the problem of a partly payed out load impacting the ground at or near maximum descent speed. The load could conceivably be a human, fragile cargo, or durable cargo. The maximum descent rate which has been selected would be roughly equivalent to a one-foot free fall, which would not be a problem for durable cargo. It is not anticipated that injured men would be lowered on the line, and a trained, able-bodied soldier would be subjected to only the equivalent of jumping off a one-foot step. Furthermore, in the case of clear visibility the man being lowered can use an arm signal to decelerate in close proximity to the ground. Fragile cargo should be either packaged to attenuate shock, or lowered to a man on the ground who has radio communication with the aircraft. This man on the ground can indirectly command deceleration at the proper moment.

Deceleration at End of Line

The lowering of a load until the end of the line is reached is a special case. Not only will a limit switch signal the stopping of the reel (brake on – descent drive off) before the end of the line is payed out, but a transducer will signal for an automatic deceleration and stop several feet before the end of the line is reached, preventing an abrupt jerk.

Ascent Command

With the storage drum stopped, the flywheel declutched, and the hoist line either fully or partly payed out, the operator moves the control in the "ascent" direction. The brake is automatically disengaged and the clutch engaged to accelerate and lift the load. The acceleration rate is automatically controlled not to exceed 0.5 g and jerk not to exceed 0.3 g, even in the event the operator pushes the switch to full speed. These rates were established to be compatible with human comfort (11). Maximum speed will be directly proportional to flywheel speed, allowing a two-to-one variation, depending on the energy level of the flywheel at a given instant. Until this flywheel related speed is reached, speed is controlled by the ascent control unit, compensating for differences in load weights.

Mid-Ascent Problems

Although minimum hoisting time is achieved by holding the ascent control full on, problems may occur, especially in wooded areas. In the event of such a problem, the operator can manually vary the ascent rate from full on to creep, or even bring the hoist to a full stop. This technique of using a slow reel-in can be used to take up slack in a line attached to a load resting on the ground before going into full speed.

Deceleration at Full Reel-In

As the human or cargo load approaches the helicopter skid gear, full-speed operation becomes more hazardous. A transducer senses the amount of line remaining payed out and, at the proper position of the load, begins an automatic ascent deceleration program. The ascent control gradually releases the flywheel clutch and engages the brake. Speed is controlled, compensating for differences in weights of the load. A limit switch at full reel-in fully engages the brake and fully disengages the flywheel clutch.

HOIST PERFORMANCE REQUIREMENTS

A series of helicopter hoist rescue mission scenarios were defined for the purpose of establishing the helicopter hoist performance requirements. The study included the following parameters:

- Lift height
- Lift speed
- Weight per lift
- Total weight lifted per mission
- Dwell time at top and bottom of lift cycle
- Energy input to hoist from helicopter
- Flywheel energy requirements

Table 3-1 summarizes 12 of the mission scenarios considered. These were presented to the USALWL project officer and a mutual conclusion arrived at between USALWL and LMSC selecting mission scenario "D" as the typical rescue mission. This scenario consists of five 200-lb lifts followed by two 400-lb lift in rapid succession for a total lifted weight of 1,800 lb. Average lift speed is 500 ft/min.

ENERGY STORAGE REQUIREMENTS

Having established scenario D as the selected mission, energy requirements for the hoist were computed using the following assumptions:

- Drive motor output = 2 hp continuous
- Total losses from motor to flywheel (including all ancillary losses) = 1 hp maximum
- Flywheel to hoist hook efficienty = 80 percent
- Maximum hoist height = 210 ft
- Average hoist or lower rate = 8.33 ft/sec or 500 ft/min.
- Maximum load = 600 lb for one hoist per mission

Table 3-1

WORST CASE MISSION SCENARIOS

Scenario	No. of Hoist Operations				Weight Hoisted (lb)
	200 Lb	400 Lb	600 Lb	Total	
A	9	0	0	9	1,800
B	7	1	0	8	1,800
C	6	0	1	7	1,800
D	5	2	0	7	1,800
E	4	1	1	6	1,800
F	3	3	0	6	1,800
G	2	2	1	5	1,800
H	1	4	0	5	1,800
I	0	3	1	4	1,800
J	0	2	1	3	1,400
K	0	1	1	2	1,000
L	0	0	1	1	600

Notes:

1. Assume 600-lb hoist is always accomplished last.
2. Assume all hoists are 210 ft at 8.33 ft/sec average rate.
3. Energy considerations:

   Energy input to flywheel $= \frac{746 \text{ w (hr)}}{60 \text{ min.}} = 12.43$ w-hr/min.

   Hoisting energy in w-hr = Weight (height)/2,655

   Flywheel energy in w-hr $= \frac{\text{Hoisting Energy}}{0.80}$

   Maximum hoist or lowering time $= \frac{210}{500} = 0.42$ min.

- Dwell times at top and bottom are the following:

Load (lb)	Dwell Time (min.)
200	0.25
400	0.50
600	1.00

(Note: No Dwell Time is required at top prior to the first or after the last hoist)

Energy requirements are as follows:

Hoisting energy	142.37 w/hr
Flywheel energy	177.97 w/hr
Energy into flywheel during mission	122.84 w/hr
Flywheel usable energy	55.13 w/hr

Table 3-2 summarizes the energy calculations for all 12 missions.

HUMAN FACTORS CONSIDERATIONS

A cursory human factors engineering evaluation of the personnel rescue hoist currently in operational use by the Army was conducted during Phase I. The primary objective of this evaluation was to determine whether or not the hoist is designed to permit the operator to do his job with efficiency, speed, and safety. This evaluation involved: (1) operation of the equipment, (2) observation of others operating and being hoisted by the equipment, (3) interviews with active-duty Army helicopter crew members and medical personnel who have been involved in rescue operations in Southeast Asia using this equipment or other methods, and (4) reviews of hoist and UH-1H publications. It rapidly became apparent that, so far as the hoist itself is concerned, deficiencies in performance that led to the award of this contract are related to:

- Problems of design
- Problems of integration into the UH-1 family of helicopters
- Problems of application in the field

Table 3-2

FLYWHEEL ENERGY CALCULATIONS

Scenario	No. of Hoists	Mission Duration (min.)	Hoisting Energy (w-hr)	Flywheel Energy (w-hr)	Energy into Flywheel (w-hr)	Flywheel Capacity (w-hr)	Flywheel Charge Time (min.)
A	9	11.81	142.37	177.97	146.84	31.13	2.50
B	8	10.72	"	"	133.28	44.69	3.59
C	7	9.88	"	"	122.84	55.13	4.43
D	7	9.88	"	"	122.84	55.13	4.43
E	6	9.04	"	"	112.39	65.58	5.27
F	6	9.04	"	"	112.39	65.58	5.27
G	5	8.20	"	"	101.95	76.02	6.11
H	5	8.20	"	"	101.95	76.02	6.11
I	4	7.36	"	"	91.50	86.46	6.95
J	3	5.52	110.73	138.42	68.63	69.79	5.61
K	2	3.68	79.10	98.87	45.75	53.11	4.27
L	1	1.84	47.47	59.32	22.88	36.44	2.93

Note: After completion of any above listed mission scenario, an additional full performance 200-lb hoist operation can be initiated by starting to lower hook after a wait of 15 sec. Similarly, an additional full performance 400-lb hoist operation can be initiated after a wait of 1.34 min.

It also became apparent that the blame for unsatisfactory operation in the field should not rest entirely with the hoist. The hoist is only one component of the total "rescue system" that is required to affect recoveries of imperiled troops or downed airmen. Other components of this total "system" are equally responsible for unacceptable delays in extracting rescues from the ground through jungle canopy or tree lines, and for the added risks such delays impose on the helicopter, its crew, and the rescuees who may already have been taken aboard. In other words, making a better hoist is not going to solve the whole problem, even though a more durable, reliable, maintainable, and operable hoist design is required for this application. An overview of the "rescue system" concept will clarify this point and disclose what can be done to improve the personnel-related elements of the hoist itself as well as hardware-related design or performance deficiencies.

An "optimized rescue system" capable of meeting all of the Army's operational requirements in Viet Nam is shown schematically in the left-hand column of Fig. 3-1. All of the elements of an idealized system, as shown in Fig. 3-2, are represented in this concept of the point-design rescue system. These include a specially-designed aircraft; an optimized hoist subsystem integrated fully with the airframe, powerplant, crew compartment arrangement, and aircraft performance/stability envelope; optimized crew stations; trained and qualified personnel equipped with the necessary personal restraint and protective gear; adequate logistic support; and rescuees who are familiar with their responsibilities in preparing for and collaborating in their safe and expeditious rescue from the combat scene. All of the elements of this system are developed together as a matched set so they will function together in a highly efficient manner with minimal risk to personnel and equipment.

Obviously, this ideal system departs significantly from the real world of operational, economic, and time constraints. Some of these realities and constraints are listed in the center column of Fig. 3-1. They are the causes or the effects of the rescue system as it exists operationally today. This

OPTIMIZED RESCUE SYSTEM ELEMENTS	OPERATIONAL NEEDS AND ECONOMIC REALITIES	EXISTING (AD HOC) OPERATIONAL SYSTEM
Special-Purpose Air Rescue Vehicle (Point Design)	• Infrequent requirement but high priority when needed	**UH-1 Family of Utility Helicopters**
Optimized Hoist Integrated into Air Vehicle • Structure • Power • Space • Dynamics • Maintainability	• Adapt family of vehicles in existing air fleet on an as-required basis • Apply existing hoist technology to personnel recovery requirement on urgent schedule	**Personnel Hoist Kit** • Helicopter electrical power constraints • Hard points • Pigtail controls
Optimized Crew Arrangement and Crew Station Design • Internal/external visibility • Communications • Coordination • Restraints • Env. and battle protection • Space for rescuees and accommodations required • Controls/displays • Etc. • Ingress/egress	• Rescue and recovery of troops becomes additional crew task with minimal additional training time made available • OJT for handling wide spectrum of operational contingencies • Evolve "doctrine" and pilot/crew technique in combat theater	**Available Crew and Hoist Operator** • Existing interphone • Level of experience in training prior to mission • Unit doctrine • Experience of other crews, other units • Ground radio link
Trained and Fully Qualified Crews & Support Personnel		
Optimized Logistic Support System	• Cannibalize and make do in the combat theater (and Conus)	**Available Spares, Tools, Equipment & Maintenance Personnel, Facilities**
Potential Rescue Population Trained in Recovery Ops.	• Accept risks of delays in rescue operations due to unfamiliarity of troops with rescue device or devices • Jettison troops on line if necessary to save aircraft, crew, rescuees	**Rescuees or Evacuees** • Most are untrained • Usually inexperienced in extraction • Usually under pressure is not under fire

Fig. 3-1 Optimal Versus Existing System

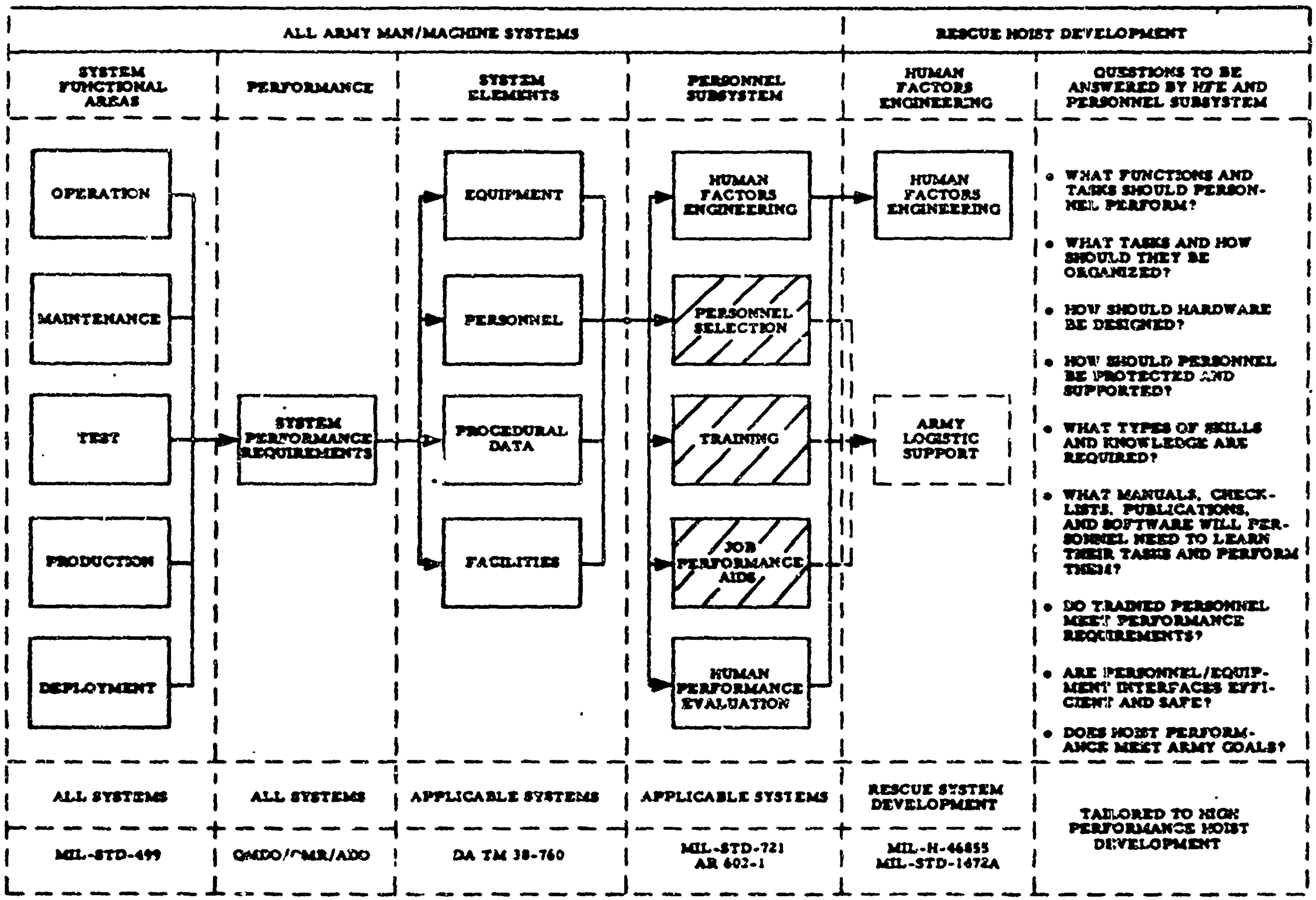


Fig. 3-2 Army Man-Machine System Development Process Applied to Rescue System and Hoist Design

existing system is simply an overlay of hardware, procedures, and doctrine, a collateral assignment, as it were, for helicopter units in-being, hastily assembled to meet an urgent but relatively infrequent need in the combat zone -- a need not sufficiently compelling to warrant the dedication of specific aircraft, crews, and equipment to do the job wherever it might have to be done within the combat radius of the assigned helicopter type. This existing system, comprising people, hardware, and the procedures for its use and maintenance, is shown schematically in the right-hand column of Fig. 3-1. When the "optimal rescue system" is compared side-by-side with the existing, it is obvious that the present assembly of components will not achieve the performance of the specially-designed system. Even though this is readily apparent, certain remedial actions can be taken to improve the system in operational use today.

Two avenues are available for improving performance. One involves improvements in the design of hardware subsystems. The second avenue available relates to the Personnel System, which the Army defines as the Human System Components and all of the aspects of design, personnel selection, training, job performance aids, and human performance testing necessary to assure that Army personnel will do their jobs accurately, rapidly, and safely, and with minimal demands upon supporting organizations. The five elements comprising the Personnel Subsystem (PS) concept are illustrated in Fig. 3-2, together with the questions these PS elements are designed to answer during the system development process. In effect, the PS concept (in use within DoD since the mid-Fifties) treats both people and hardware as system components. As with hardware components, people can be selected for their suitability to perform required functions, they can be modified through training to achieve improved performance or additional capabilities not available in the "off-the-shelf design," and they can be tested to assure satisfactory performance as part of a man-machine system. Although this is an oversimplification of the concept, the potential for improving the performance of any system lies at least in part in improving the performance of the human components. Figure 3-3 illustrates this parallel in outline form.

SYSTEM DEVELOPMENT PROCESS	HUMAN COMPONENTS	HARDWARE COMPONENTS
• What function should the component perform?	• Human capabilities and limitations	• Equipment capabilities and limitations
• How should we allocate it?	• Manual or semi-automatic	• Semi- or fully automatic
• Will existing components do the job effectively?	• Catalog of MOSs	• Catalogs of available hardware
• If not, which components require least modification?	• Select MOS with knowledge, skills, etc., closest to required	• Select hardware requiring least modification from standard
• Modify and verify performance	• Train to achieve required knowledge, skills, etc., provide required personal equipment; and test to verify performance.	• Modify by design to achieve required performance and interfaces with other components and environment: test to verify performance.
• If a new design is required, what are design and performance requirements?	• "Qualitative and quantitative personnel requirements," MOS description, selection criteria, training and training equipment	• End item design and performance specification
• What are integration requirements?	• Man-machine and environmental protection requirements, communication requirements, etc.	• Interface design and performance requirements
• Will the new or modified component meet functional performance goals?	• "Personnel Subsystem Test and Evaluation" (PST&E)	• Functional testing, reliability testing and verification, maintainability demonstration, PST&E

Fig. 3-3 Personnel Subsystem: People, Like Hardware, Are System Components

Since improvements in system performance can be achieved by modifying either the hardware or the human components or both, several degrees of freedom are available in selecting remedial actions. Figure 3-4 traces a path through a series of possible modifications in the Personnel Subsystem and hardware subsystems, any one of which has the potential of improving performance. Note that most of the PS modifications involve: (1) increasing the knowledge or skills of the rescuees by training, briefing, better printed (or painted) instructions, or voice-communicated instructions, or (2) increasing hoist operator performance by improving his knowledge and skills by training, practice in operations, and improved job performance aids. Since human factors engineering is the hardware-oriented PS element, improvements in the design of the rescue device and the hoist and its related components can also enhance rescuee and operator performance. Some of these remedial actions are, perhaps, beyond the scope of the USALWL mission. Most that are directly related to the design of the hardware and the procedures for its use are not. The main thrust of this and future phases should therefore concentrate on these remedial actions until such time as an optimal system is established as a firm requirement by the Army.

Problem of Design

As stated above, evaluations of the existing hoist and the procedures for its use disclosed several operability problems related to hardware design. The most significant of these are listed below:

1. Manual hoist controls incorporated into the control handle/cable assembly are designed for left-hand operation. Since the left hand is frequently used to guide or to provide tension on the cable, hoist controls are frequently operated by the right hand, just the opposite of what they were designed for.
2. Control "feel" (feedback) in the two-speed range control swtich is poor. The "breakout" force to switch from OFF to SLOW, and from SLOW to FAST is high, making "inching" (small movements of the cable) difficult.

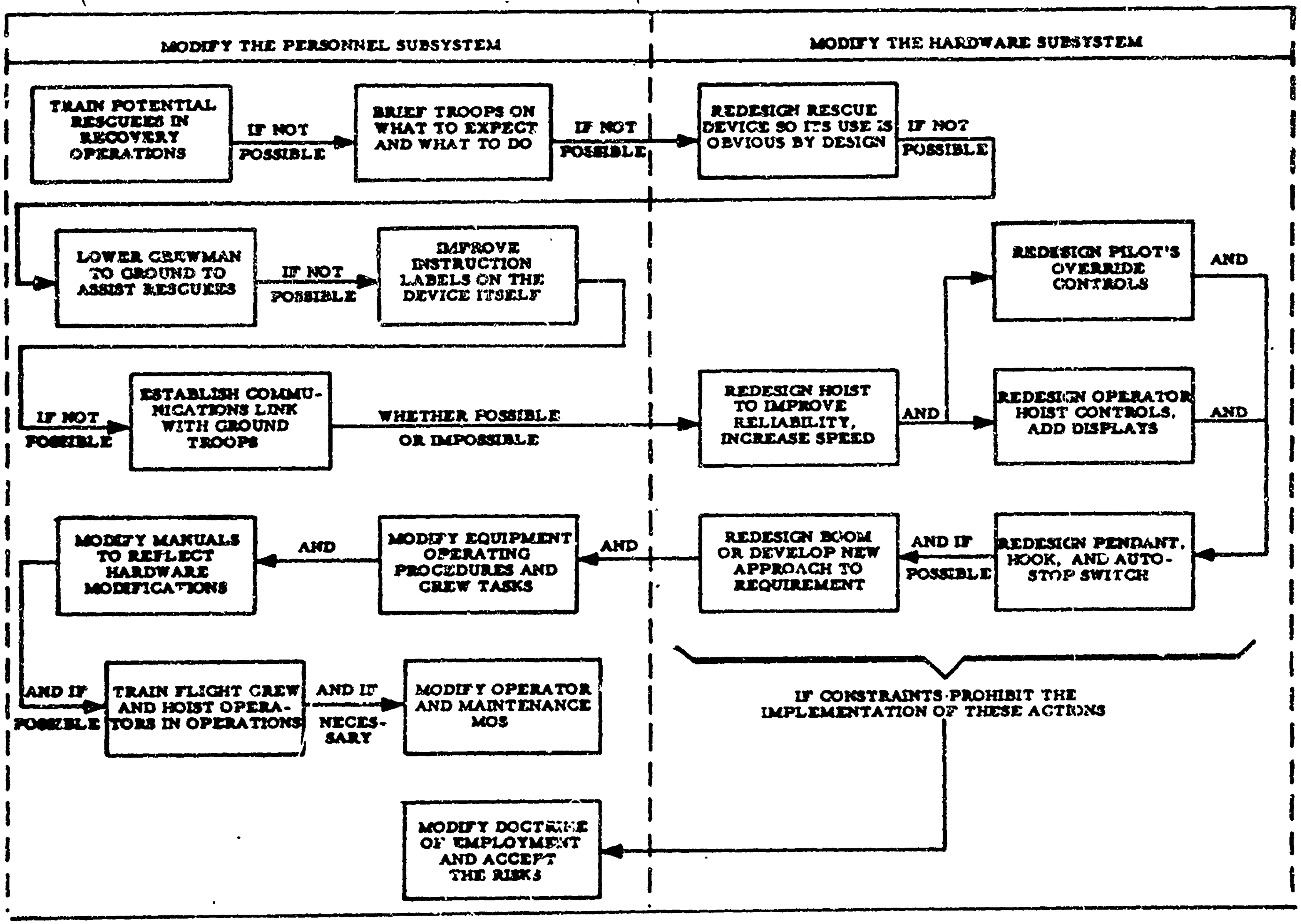


Fig. 3-4 What Can Be Done to Improve System Performance?
(Degrees of Freedom Available in Selecting Remedial Actions)

3. Control "feel" in the intercom switch is likewise poor, but in this case because it is soft.
4. There is no direct indication of the length of cable payed out.
5. The pyrotechnic or ballistic cable cutter is unreliable. This is a Flight Safety requirement; aircraft have been lost because a jungle penetrator or other recovery device has caught in trees and neither the hoist operator nor the pilot was able to cut the cable to prevent upsetting.
6. Exposed hoist mechanisms are potentially hazardous and visually distracting during operations. They are easy to damage accidentally or willfully, if an operator wishes to abort a mission.
7. The need to provide tension on the cable and to control the hoist simultaneously forces the hoist operator to lean way out of the compartment. Poor downward visibility usually requires him to do the same thing. This is a stressful operation at best, requiring ultimate confidence in the restraint harness and safety strap assembly that connects him to aircraft structure. The hoist itself does not offer an obvious and convenient hand hold for hanging on.
8. The hoist boom swings into the area needed by the operator to man-handle rescuees into the aircraft.

Problems of Integration

Certain problems associated with hoist operation have arisen because of the requirement to install the unit in the UH-1 family of helicopters. These are referred to as problems of integration:

1. Installation and assembly of the hoist by one man in one hour is desired by TM 55-1520-210-10, Operator's Manual, Army Model UH-1D/H Helicopters, and UH-1-24, Installation and Operating Instructions - Internal Rescue Hoist, Revision A, dated 14 May 1971. The hoist assembly exceeds the maximum weight specified for a one-man lift in MIL-STD-1472A, Table X, however.

2. The tasks of the hoist operator must be performed in squatting, stooping, kneeling, or prone positions due to the limited internal height of the UH-1D/H cabin. All of these working positions are inefficient for man-handling rescuees, and lengthen the time required for installing the hoist and performing personnel rescue/recovery operations.
3. Instructions for installing the hoist are not simple or particularly lucid. Frequent installation and removal as required by combat needs does decrease the time required to perform these tasks because of learning and practice effects. Reductions in performance times could be achieved at the outset, however, by utilizing PIMO-formatted operation and maintenance manuals instead of conventional TM-formatted manuals.
4. The UH-1 skid-type landing gear common to this family of helicopters interferes with downward visibility, diminishes penetrator clearance as the cable is payed out and retrieved, interferes with the rescuee as he is being brought up to the aircraft, and lengthens the time devoted to positioning an able-bodied man for retrieval into the aircraft cabin. The skid gear seriously impedes retrieval of injured or disabled rescuees, requiring careful positioning and rotating on the part of the operator to prevent the rescuee from being further injured by the skid gear; this, of course, introduces further delay and lengthens turnaround time.
5. The length of the hook and attached rescue device (jungle penetrator, sling, or other device) is such that the rescuee is below the level of the cabin deck when the cable is fully raised. This also results in delays in positioning the rescuee for retrieval (pulling/dragging) into the aircraft.
6. The jungle penetrator is an innovative concept and apparently the best device currently available for extraction of rescuees through tree-covered ground. Its design is not optimized for deployment and mounting without assistance, however, and its appearance is puzzling to the untrained user. The longest delay in a typical mis-

sion timeline is due to the time necessary to figure out what to do with it, how to mount it, and how to restrain yourself on the way up. The time required is lengthened for a two-man lift, even though such lifts are performed infrequently. Three-man lifts are never tried with the penetrator, even though it was designed for that maximum capability.

Problems of Application

Certain problems have arisen as a result of the manner in which the hoist has been used in the field, or rather, mis-applied. As suggested above under problems of design, the hoist is vulnerable to willful negligence or intentional destruction through jamming, over-temperature operation leading to motor failure, and intentional breakage because vital components are exposed. While this type of damage cannot be entirely eliminated, redesign of the hoist and incorporation of protective covers can result in significant reductions in downtime and improvements in equipment availability when rescue missions must be performed.

The performance of personnel rescue or extraction missions is hazardous under the best conditions in a non-hostile environment. Under combat conditions, dwell time in hovering flight often means flirting with death. Reducing the time required in hover for a given lift is therefore a critical factor in survivability. The lack of hoist reliability and the widespread practice of limiting lifts to two men as a maximum have led to a condition of "under using" the hoist. In other words, the full capacity of the hoist/aircraft is rarely utilized during any one ground-to-aircraft lift cycle, thus increasing the number of cycles and the time required to rescue a given number of men. Overloading the hoist/aircraft during lifts is very rare because of the upsetting moment imparted to the vehicle during the lift. The tendency, then, is to underload. The only way of knowing what impact a load will have on an aircraft is to apply collective pitch to raise the load off the ground before hoisting in order to determine whether the limits of aircraft controllability

are being approached. There is, however, no way of knowing whether the limit of hoisting capacity is being approached Moreover, there is no way of knowing whether the hoist is reaching its limiting temperature; the operator simply has to comply with the cautions set down in the applicable instructions in order to avoid an over-temperature condition. Once the limited number of consecutive lifts has been performed, the helicopter must fly around until the hoist motor has cooled. Lack of direct readings of capacity remaining and of temperature, then, results in further delays and the expenditure of precious time and fuel in cooling the unit before continuing the rescue/extraction mission.

Human Factors Consideration in the High Performance Hoist

While it is recognized that the development of an optimized "rescue system" (Fig. 3-1) is beyond the scope of this effort and exceeds the mission of the USALWL, significant strides can be made in the achievement of improved performance of existing system elements, including the hoist, the operator's controls and displays, the pendant hook, the jungle penetrator, and the related personnel subsystem elements that do fall within the responsibility of the procuring activity. Many of the problems of design, integration, and application reported above can be minimized and some completely eliminated through the prudent application of the systems approach and related human factors engineering principles during the development of the high performance hoist concept.

Of the two proposed designs, the preferred concept shown in Fig. 7-21, Sect. 7, was evolved on the basis of a systems approach. It incorporates innovations that eliminate the sources of some of the problems enumerated above. Placement of the hoist and the extension boom at opposite sides of the helicopter cabin results in a more favorable distribution of weight about the longitudinal centerlines of the aircraft and permits the boom and integral pulley to reach farther out of the cabin than with either the existing or the new boom-mounted units. Clearance past the skid gear is significantly increased; this simplifies

the task of raising the rescuee up to the threshold of the cabin and drawing him inside. The hoist and drive unit no longer consume valuable space in the primary ingress opening. Their separation into two units for installation in the aircraft results in smaller, lighter units that should be easier for one man to install, one at a time. This configuration should also be more versatile than the existing design, permitting installation in a variety of helicopters, not just the UH-1 family. As shown, the basic design of this unit will essentially eliminate design problems 6 and 8 and integration problems 1, 4, and 5, noted above.

An electromechanical guillotine device for severing the cable and a simple guarded mechanical backup are suggested as alternatives to the existing gas-operated device to increase reliability to an acceptable level. This effort will address design problems 1 through 5, and result in recommendations for reconfiguring the pilot's hoist controls as well.

As indicated in the discussion of personnel subsystem modifications to improve overall system performance, attention should be paid to the preparation of installation and operating instructions for the redesigned hoist, including material for insertion in the UH-1D/H -10 Technical Manual, and to the preparation of supplemental training and familiarization requirements. Adoption of PIMO-formatted operating and maintenance manuals may be premature during Phases I and II of the high performance hoist development effort; therefore, it is recommended that their preparation be deferred until the assignment of product responsibility is made to a commodity command. In the interim, preliminary operating instructions will be provided.

Increasing the rate of descent and ascent from 100 fpm to 500 fpm will not result in significant acceleration stress upon the crewman descending with the hoist or the rescuee ascending on a jungle penetrator. Accelerations will be experienced in the plus G_z direction (eyeballs down); tolerance in

this direction along this axis far exceeds the rates of onset and levels that will be experienced during hoisting operations.

Redesign of the hook to prevent accidental release of a rescue device will be required. In addition, it would be very desirable to review the design of the jungle penetrator in detail, and to determine the ways in which that device can be reconfigured to assure that its use is obvious by design. If significant improvements in "boarding time" cannot be achieved over those experienced in the field, alternative designs will be explored and the most promising configuration will be recommended for further development.

Section 4
FLYWHEEL STUDY

Better materials for fabricating flywheels and the use of computer analysis in determining optimized flywheel shapes and operating speeds has in recent years greatly advanced flywheel technology. Its use as a highly effective energy storage device has enhanced its suitability for many applications including passenger automobile and mass transit vehicles. Its ability to store more energy per pound than almost any other form of energy storage system makes it a strong contender for use on board a helicopter where any form of energy is at a premium.

FLYWHEEL DESIGN STUDY

Energy is put into a flywheel by increasing its rotational speed, thereby imparting an increase in kinetic energy to the rotating mass of the flywheel. This stored energy is used by the load connected, causing the flywheel to slow down. Common practical applications of the flywheel are seen in the internal combustion engine and the inertia aircraft starter.

The specific energy of a flywheel is represented by the simple relationship:

$$E = K_S \frac{\sigma}{\rho}$$

where:

- E is specific energy in in.-lb/lb
- K_S is the flywheel shape factor (dimensionless)
- σ is the material design working stress level in psi
- ρ is the material density in lb/in.^{3}

FLYWHEEL SHAPE

Flywheel shape factors for several flywheel geometries are shown in Table 4-1.

The superior energy density of the constant-stress flywheel may be attributed to the fact that all particles of the flywheel are placed in bi-axial stress to a uniform predetermined working level. By contrast, in the rim-type flywheel, only the outer band of particles of the rim are at the maximum predetermined working stress level and this band is only in uni-axial hoop stress. An obvious choice for the helicopter hoist flywheel would be a tapered disc of exponential or conical cross-section.

Table 4-1
FLYWHEEL SHAPE FACTORS FOR VARIOUS GEOMETRIES

Flywheel Geometry	Shape Factor (K_S)
Constant-Stress Disc (OD $\rightarrow \infty$)	1.00
Modified Constant-Stress Disc (typical)	0.931
Truncated Conical Disc	0.806
Flat Unpierced Disc	0.606
Thin Rim ($\frac{ID}{OD} \rightarrow \infty$)	0.500
Shaped Bar (OD $\rightarrow \infty$)	0.500
Rim with Web (typical)	0.400
Single Filament (about transverse axis)	0.333
Flat Pierced Disc	0.305

In order to optimize the design of the flywheel, tradeoffs must be made involving flywheel speed, diameter, minimum thickness, maximum stress, and weight. In order to facilitate this process, various existing LMSC flywheel designs were rescaled with respect to speed and diameter to give the resultant minimum thickness, maximum stress, and weight. The use of some type of steel was assumed since previous studies have shown that only steel has the necessary high strength to density ratio for good energy density and relatively high density for minimum diameter. A programmable calculator routine was written for this purpose. The calculator outputs were arranged in matrix form to facilitate evaluation. A typical set of output is shown in Fig. 4-1. The flywheel scaled in Fig. 4-1 is a constant stress exponential disc with a rather large hub-to-tip thickness of 7.12, giving a shape factor of 0.93. An examination of the results in Fig. 4-1 shows the maximum stress levels to be very low.

It is possible to reduce the flywheel weights shown in Fig. 4-1 by using an exponential flywheel with a lower hub-to-tip thickness, but this will also cause an increase in maximum stress. Figure 4-2 shows the same matrix using a "flatter" exponential disc with a hub-to-tip thickness of 2.17, giving a shape factor of 0.81. Comparing results of the two matrices, it is seen that the flatter disc (Fig. 4-2) has lower weights and higher stresses. Because the stress values are so low in either case, the flatter shape was chosen because of its lower weight. Further flattening (toward a flat disc) does not appear warranted as the stresses increase faster than the weight goes down.

In selecting the final speed and diameter, tradeoffs were made among speed, diameter, machinability (minimum thickness), bearing life, bearing losses, envelope dimensions, safety, gearing requirements, and weight. The speed and diameter finally chosen were 28,000 rpm and 12-in. diameter.

Diameter, in.	11 S	12 S	13 S	14 S
Speed, krpm	26 S	26 S	26 S	26 S
Min. thickness, in. . .	0·386708 A0	0·273041 A0	0·198234 A0	0·147380 A0
Stress, psi . . .	28028·269465 A0	33356·199360 A0	39147·132750 A0	45401·420130 A0
Weight, lb	25·624768 A0	21·531782 A0	18·346641 A0	15·819293 A0
	11 S	12 S	13 S	14 S
	27 S	27 S	27 S	27 S
	0·358593 A0	0·253191 A0	0·183822 A0	0·136665 A0
	30226·081215 A0	35971·406295 A0	42216·439815 A0	48961·181775 A0
	23·761547 A0	19·966371 A0	17·012766 A0	14·669139 A0
	11 S	12 S	13 S	14 S
	28 S	28 S	28 S	28 S
	0·333437 A0	0·235428 A0	0·170926 A0	0·127078 A0
	32506·496715 A0	38685·100740 A0	45401·420130 A0	52655·124390 A0
	22·094611 A0	18·565763 A0	15·819293 A0	13·640048 A0
	11 S	12 S	13 S	14 S
	29 S	29 S	29 S	29 S
	0·310837 A0	0·219472 A0	0·159341 A0	0·118465 A0
	34869·535965 A0	41497·613190 A0	48702·404190 A0	56483·247975 A0
	20·597304 A0	17·307462 A0	14·747082 A0	12·715599 A0
	11 S	12 S	13 S	14 S
	30 S	30 S	30 S	30 S
	0·290460 A0	0·205084 A0	0·148896 A0	0·110699 A0
	37316·190450 A0	44409·274140 A0	52119·391995 A0	60446·213520 A0
	19·246831 A0	16·172712 A0	13·780252 A0	11·881942 A0

Fig. 4-1 Flywheel Diameter-Speed Matrix, Exponential Disc, Hub-to-Tip Thickness = 7.12

Diameter, in.	11 S	12 S	13 S	14 S
Speed, krpm	26 S	26 S	26 S	26 S
Min. Thickness, in. .	0.563170 A0	0.397635 A0	0.288693 A0	0.214633 A0
Stress, psi . . .	44135.342496 A0	52524.664316 A0	61643.650552 A0	71492.041352 A0
Weight, lb	18.632262 A0	15.656286 A0	13.340241 A0	11.502558 A0
	11 S	12 S	13 S	14 S
	27 S	27 S	27 S	27 S
	0.522226 A0	0.368726 A0	0.267704 A0	0.199029 A0
	47595.791580 A0	56642.798812 A0	66476.767826 A0	77097.308844 A0
	17.277607 A0	14.518017 A0	12.370357 A0	10.666273 A0
	11 S	12 S	13 S	14 S
	28 S	28 S	28 S	28 S
	0.485590 A0	0.342859 A0	0.248924 A0	0.185066 A0
	51186.556442 A0	60916.194878 A0	71492.041352 A0	82913.836012 A0
	16.065570 A0	13.499549 A0	11.502558 A0	9.918023 A0
	11 S	12 S	13 S	14 S
	29 S	29 S	29 S	29 S
	0.452679 A0	0.319621 A0	0.232053 A0	0.172523 A0
	54908.026860 A0	65344.982440 A0	76689.860908 A0	88942.142560 A0
	14.976702 A0	12.584610 A0	10.722946 A0	9.245800 A0
	11 S	12 S	13 S	14 S
	30 S	30 S	30 S	30 S
	0.423004 A0	0.298669 A0	0.216841 A0	0.161213 A0
	38760.202834 A0	69929.421350 A0	82070.096568 A0	95181.838710 A0
	13.99486[illegible] A0	11.759591 A0	10.019988 A0	8.639686 A0

Fig. 4-2 Flywheel Diameter-Speed Matrix, Exponential Disc, Hub-to-Tip Thickness = 2.17

FLYWHEEL MATERIAL

The maximum stress in the flywheel is seen from Fig. 4-2 to be 60,916 psi. This stress level is sufficiently low for a number of high strength steels to be suitable. In order to maximize safety factors, the alloy chosen was 4340. This alloy can be heat treated as high as 260,000 psi while maintaining good ductility and elongation, permitting safety factors as high as 4:1.

GYRODYNAMICS

The most obvious and useful characteristic of a flywheel is its ability to absorb, store, and release rotational energy. A secondary characteristic is the precessional torque effect which is detrimental in the case of the helicopter hoist. Precession torque is developed by the flywheel and imposes relatively high radial loads to the flywheel support bearings whenever a disturbing moment acts on the flywheel in a direction normal to the flywheel spin axis. This gyrodynamics effect depends upon a special form of Newton's law which states that a rapidly spinning body rigidly resists being disturbed and tends to react to a disturbing torque by precessing in a direction right angles to the disturbing torque. The upsetting torque is imposed by rotational movements of the helicopter. Roll, pitch, and yaw motions are common to helicopter flight and these motions could easily impose the disturbing torque upon the flywheel unless the flywheel was suspended in a gimbal or other mount which could not transmit the disturbing torque.

The magnitude of the precession torque is calculated as follows:

$$T_P = I\omega\Omega$$

where:

T_P = precession torque, ft-lb

I = flywheel inertia, ft-lb/sec^2

ω = flywheel rotational speed about its spin axis, rad/sec

Ω = rotational speed normal to spin axis, rad/sec

Discussions with the stability and controls group engineer at Bell Helicopter regarding roll, pitch, and yaw rates of the UH-1H helicopter established that a 1.0 rad/sec rotational speed was not uncommon and is the rate normally used in analysis involving extreme helicopter maneuvers.

Although the bearing loads as a result of precession torque loads are relatively high, the system does not require special suspensions or gimbal mounts. The cumulative effect of these short duration loads are not sufficient to cause any great harm to or seriously reduce the life of the flywheel bearings. (Reference Appendix A2.)

The calculated precession torque for the selected flywheel at rated flywheel speed and 1.0 rad/sec disturbing rotation rate is 132 ft-lb which is resisted at each support bearing by a radial force of 396 lb. The effect upon helicopter control is minimal.

A more detailed analysis of these precession forces are included in Appendix A2.

Section 5
SAFETY AND COST CONSIDERATIONS

SAFETY CONSIDERATIONS

The flywheel powered hoist is primarily intended to alleviate some of the hazards presently associated with helicopter hoist rescue operations due to a shortening of hoisting and hovering time. It is necessary to be assured that the addition of the flywheel does not introduce significant new hazards which would offset this advantage (see Fig. 5-1). All forms of stored energy can be hazardous if the energy is released in an undesired manner. Furthermore, the flywheel is also a gyroscope and as such can transfer precessional forces into the airframe. High precessional forces could adversely affect the stability and control of the aircraft.

Investigation of the gyroscopic effects of flywheel hoist installed in a UH-1 helicopter indicates that these effects are not significant. With a fuselage pitch rate of one radian per second, a roll moment of 132-ft-lb is generated. This is equivalent to a 132-lb man moving one foot laterally which is well within the controllability of the aircraft.

Undesirable release of energy stored in the flywheel can be in the form of: catastrophic disintegration, transfer of moment into the airframe, excessive heating, or excessive vibration. Catastrophic disintegration, in turn, could conceivably be the result of overspeed, fatigue failure, deficient physical properties or ballistic or crash impact.

The hoist system under consideration is not in any danger of critical overspeed. The helicopter generator itself maintains a maximum operating speed due to the requirement of not exceeding the maximum rotor design speed beyond an established margin. Furthermore, voltage protection can be provided in case of voltage regulator failure. The hoist motor maximum speed

HAZARD	Resolution
GYROSCOPIC MOMENT	• Less than man moving one foot
OVERSPEED	• Beyond motor maximum speed • Flywheel disengaged for descent mode
FATIGUE	• Less than 10,000 stress cycles • 4 to 1 safety factor
DEFICIENT PROPERTIES	• Overspeed testing
BALLISTIC IMPACT	• Small presented target area
CRASH IMPACT	• Protected location • 4 to 1 safety factor
MOMENTUM TRANSFER	• Opposite to load imposed moment • Automatic jettisoning
FRICTION HEATING	• Operational history
CRITICAL VIBRATION	• Attenuated by installation mounting • Jettisoning

Fig. 5-1 Flywheel Safety Aspects

selection is well below the burst speed of the flywheel, thus ensuring against overspeed and possible bursting of the flywheel. The design decision to disengage the flywheel from the drum during the descent phase assures that energy from the descending load cannot add speed to the flywheel.

A potential fatigue failure problem could exist if the flywheel were subjected to sufficient stress cycles resulting from repeatedly spinning the flywheel from rest to full speed. To assure that this does not occur, a design criteria of 10,000 stress cycles is established (equivalent to an annual average of 1,000 full cycles for a ten-year life). Since consecutive lifts of 200-lb do not constitute full cycles, this assumption seems to be conservative. The stress level of the selected flywheel is 61,000 psi at a design speed of 28,000 rpm. Material selection and hardening can readily provide a 4-to-1 safety margin.

A flywheel could also fail if its physical properties are less than specified. To assure that this does not occur in the manufacturing process an overspeed spin test can be run on each flywheel, over and above normal quality control procedures. Furthermore, precautions must be taken to prevent deterioration such as corrosion during storage. Finally, inadequate maintenance should be guarded against. Unit replacement with depot overhaul of the flywheel chamber is recommended to avoid maintenance deterioration.

Severe impact on a flywheel spinning at full speed must be considered. The only impacts which can be conceived as having sufficient magnitude to destroy a spinning flywheel are crash landing and a direct hit by a projectile. The results of a ballistic impact upon a fully stressed flywheel are not known at this time. Such a hit may be radial, axial, or oblique. Even if it is eventually established that the prob [illegible] of the flywheel bursting is relatively high, the probability of the [illegible] is relatively low. The total circular area of the flywheel is less [illegible] much smaller than vital targets within the helicopter such as the flight [illegible] pilot, engine, and transmission. It is unlikely that this full circular target will be presented. It

is much smaller in edge view or obliquity. Furthermore, adjacent equipment such as the electric motor and hoist drum can provide shielding from small arms fire.

The possibility of a flywheel disintegration greatly increasing the radius of destruction in a fatal crash requires that adequate crash load criteria be assigned to the flywheel assembly. It is fortunate that the flywheel occupies the safest location in the aircraft in respect to landing impact. It is most vulnerable in the event that the helicopter falls from a considerable height and lands flat on the side of the fuselage on which the flywheel is installed. Otherwise, the full ground impact is greatly attenuated by progressive deformation of the landing gear, fuselage pan, cabin structure, mounting structure, and flywheel enclosure. The flywheel at full design speed experiences 140,000 g's at the outer rim. Current vertical crash load criteria requires that helicopter installed equipment hold together at 45 g's. If a 4/1 stress safety factor is used, the flywheel will burst at 560,000 g's applied in a radial direction at the outer rim.

If a significant portion of the momentum stored in the flywheel is suddenly transferred into the airframe, an uncontrollable moment could result. With the selected configuration this would be a roll moment. The momentum transfer could result from a bearing, gear train, or vacuum failure, or from a distorted housing. It is anticipated that in each case the transfer would be over a considerable length of time. However, during the critical portion of the hoist operation, the pilot may be using all available lateral cyclic pitch. To avoid additional moment being applied the flywheel is installed so that any roll moment would be in the opposite direction from that imposed by the suspended load. It is not anticipated that even with a light suspended load the roll moment imposed by a faulty hoist assembly in the opposite direction would exceed the control available to the pilot. However, if future investigation proves that this is indeed a significant problem, automatic emergency jettisoning of the flywheel assembly is a distinct possibility. In this event the moment imposed by the flywheel can shear a section at the top mounting and rotate outboard on a pivoted bottom mounting.

A flywheel can generate heat by either rubbing on its housing or by a sudden vacuum failure. In the case of friction heating with contact with the housing an ignitable substance is not present. However, in the case of a vacuum failure, both combustible oil and air might be subjected to heating. Fortunately, there is some operational experience in this area. The following quotation is from Reference 9.

> "The flywheel tip is traveling at 132,000 feet per minute (mach 2.3) in an evacuated housing (2 cm Hg abs). It was originally feared that if a leak occurred and a lubricating oil - air mixture entered the housing, there was danger of a fire or explosion. Several leaks of this type have occurred and, so far, the only effect has been flywheel slowdown and increases in flywheel housing temperatures approaching 350°F . . . . "

This indicates that is possible to design a flywheel and its assembly so that ignition temperature is not achieved.

An unbalanced flywheel can produce vibrations varying in magnitude with the degree of unbalance and varying in frequency from zero up to order of 500 Hz. All critical frequencies in between will be passed through as the flywheel slows down. Much of this vibration will be attenuated by the installation mounting. However, it is anticipated that out-of-balance problems can be avoided by design, quality control, and maintenance procedures.

COST CONSIDERATIONS

Cost considerations involved in introducing a new high performance rescue hoist include various aspects of the total life-cycle cost of the hoist, and the effect of the new hoist upon the cost-effectiveness of the total air rescue system. Life-cycle costs include all development, procurement, and operating expenditures, including impact upon the Army's logistics system, over the life of product from the "cradle to the grave."

Life-Cycle Costs (Figure 5-2)

A new, high performance hoist will have certain non-recurring start-up costs including prototype engineering, test and evaluation, production engineering, documentation and tooling, service and operation manuals, initial spares, and introduction of a new line item. The relative significance of these costs in comparison to the total life-cycle cost is dependent upon the time duration of the life cycle. If this equipment is tied to one type of aircraft, which will be phased out in the not-too-distant future, these costs will be relatively high. To avoid this situation a design goal has been established to make the hoist compatible with a wide variety of configurations, so that it will not become obsolete with the introduction of the next generation of helicopters.

Initial acquisition cost varies with average unit cost and the number of units required to supply a selected number of aircraft. The average unit cost is sensitive to complexity, weight, production quantity, and production rate. It is not anticipated that there will be a reduction in complexity or weight with the introduction of a new, higher performance hoist. However, a design goal for the new hoist is to provide sufficient versatility for the hoist to be applicable to more than one type of aircraft, so that average unit cost advantages can be obtained.

The number of units required to supply a given size of helicopter fleet may well vary between different hoists, depending on relative operational availability. If hoist availability is low, as in the case of the existing equipment, a large maintenance float quantity is required to keep the desired number of aircraft equipped. The new hoist concept has unloaded the electric motor increasing its reliability and eliminated the fatigue problem of a wire rope running through a capstan. The flywheel itself is a long life, high reliability component. Hoist availability will be improved if the new design incorporates field drum replacement, which will mean that the hoist will no longer be sent to a rear echelon to change a hoist cable.

TYPE OF COST	Approach
DEVELOPMENT	Extended service life through compatibility with next generation aircraft
AVERAGE UNIT	Improved production quantity and rate through applicability to a variety of aircraft
ACQUISITION	Reduced maintenance float requirements through improved hoist availability
TRAINING	Increased lift cycles per training flight hour
OPERATING	- Increased reliability - Easier hoist line replacement - Lower hoist line cost

Fig. 5-2 Life Cycle Costs

Increased motor reliability, increased hoisting line life, and field replacement of hoisting line also have operating cost advantages reflected in reduced maintenance manhours and reduced consumption of spare parts. Furthermore, if synthetic rope is used in place of the present stainless steel wire rope, the cost of a replacement hoist line can be reduced to a half.

Helicopter rescue hoist operational training costs can be very significant since considerable costly flight time is required. The costs of a pilot, copilot, and instructor must be included, as well as that of the students. The number of hoist lift cycles experienced by the student is indicative of his degree of training, and more lift cycles per flight hour are possible with the substitution of a high performance hoist for present equipment. Furthermore, the number of students aboard the helicopter, observing each others performance can be profitably increased, if each has the opportunity to operate the hoist himself during the instruction session.

Cost Effectiveness (Figure 5-3)

The cost effectiveness of the rescue hoist must be viewed as part of the cost effectiveness of the total rescue system. The measure of effectiveness for this total system could be expressed as "dollars per man rescued."

The high performance hoist has a five times increase in speed and does not require a cooling off period between cycles. Thus, for a given flight time more men can be evacuated. Conversely, less flight time is required to rescue a given number of men. Dollars per flight hour can be established for any particular aircraft.

However, the primary purpose of high performance from a rescue hoist is to evacuate more men, while spending less time hovering. Extended hover time means increased vulnerability due to engine failure and ground fire. Not only must the cost of helicopters lost be added to the total cost of rescue operations, but any helicopter crewme[illegible] must be subtracted from the total men rescued.

$/MAN RESCUED

- MORE MEN RESCUED PER FLIGHT HOUR
- LESS FLIGHT HOUR PER MAN RESCUED
- LESS HOVER HOURS PER MAN RESCUED
- LESS AIRCRAFT LOST PER MAN RESCUED
- LESS CREWMEN LOST PER MAN RESCUED
- FEWER MISSIONS ABORTED
- MORE HOIST CYCLES WHEN REQUIRED

Fig. 5-3 Cost Effectiveness

In addition to an increase in hoist speed, the selected hoist design has the potential capability of in-flight replacement of a drum and its hoisting line. This feature could eliminate a portion of mission aborts resulting from a requirement to use the guillotine. Being able to complete the rescue on the same mission with a new line obviously pregents a significant increase in dollars per man rescued.

Cost effectiveness of the rescue system is drastically affected by aircraft availability rate, which is in turn affected by the availability of the hoist. Thus, the planned improvement in hoist availability is sifnigicant in terms of cost effectiveness as well as life-cycle cost.

Section 6
DESIGN STUDIES

FLYWHEEL SPIN-UP TECHNIQUES

Only a limited supply of electrical power is reserved for hoist operation during any rescue mission. The present hoist uses a maximum 117 amp from the helicopter 28 V d.c. supply. Since the proposed high performance hoist is constrained to work to this same maximum power consumption or less, the flywheel spin-up technique used must be efficient.

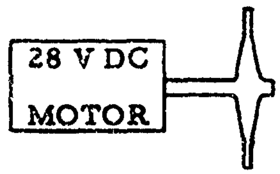
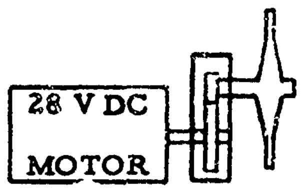
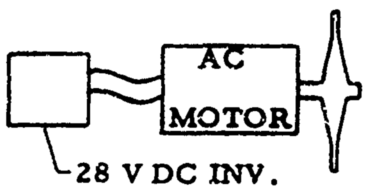
The inefficiencies of utilizing a motor to drive a hydraulic or pneumatic pump and, in turn, using the pump flow and pressure to drive the flywheel with hydraulic or pneumatic motors is obvious. The tradeoff study then becomes simply a comparison of motor parameters in connection with associated requirements such as inverters or connecting mechanical drive systems. Table 6-1 briefly summarizes three possible ways of spinning up the flywheel.

The advantages of a system coupling the motor shaft directly to the flywheel shaft are many.. High speed with no gear system promises low weight and simplicity. Unfortunately, commutative problems limit motor speeds to about 18,000 rpm, well below optimum flywheel operating speeds. The a.c. motors requiring no commutation could meet the high speed characteristics required of a direct flywheel shaft connection but it requires an expensive, complicated inverter for changing the helicopter electrical system d.c. current into a usable a.c. current.

The best system shown as number 2 in the table incorporates a step-up gear ratio to permit the flywheel and motor to both operate at its own optimized speed.

Table 6-1

FLYWHEEL SPIN-UP TECHNIQUES

COMPONENT ARRANGEMENT	CONFIGURATION SCHEMATIC	STATE-OF-THE-ART DESIGN	EFF.
HIGH SPEED DC MOTOR WITH DIRECT DRIVE		NO, OPTIMIZED FLYWHEEL SPEED TOO HIGH FOR EFFECTIVE COMMUTATION	–
DC MOTOR WITH STEP-UP GEAR RATIO		YES, EASIEST, MOST DIRECT METHOD	GOOD
INVERTER WITH AC MOTOR, DIRECT DRIVE		YES, POOR EFFICIENCY IN CONVERTING DC TO AC	POOR

FLYWHEEL/HOIST DRIVES

A detailed trade study considering many types of drive systems for connecting the flywheel powered shaft to the hoist mechanism was conducted. The most obvious results of this study establishes the fact that commercially available variable drive units in the 10 - 20 hp category are limited to maximum output speeds of 3,500 to 8,000 rpm and package weight was generally too heavy for serious consideration for the lightweight, high performance hoist requirements. Table 6-2 is a brief summary of the trade study of the drive systems considered for the hoist transmission.

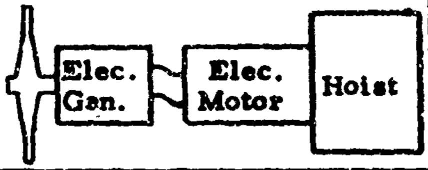
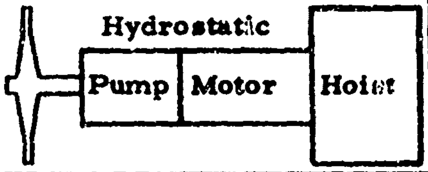
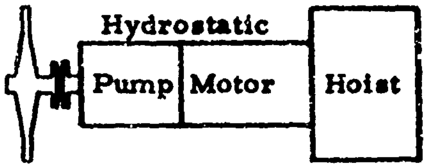
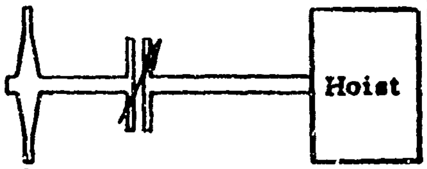
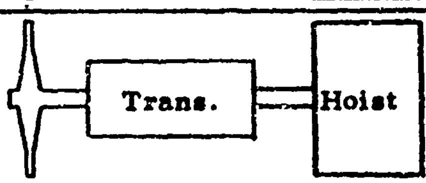
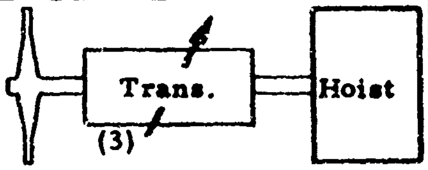
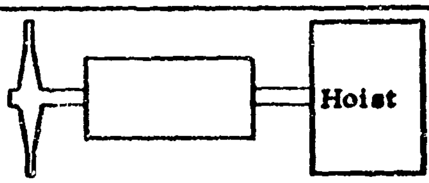
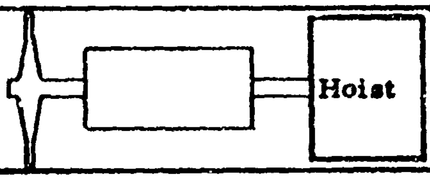
Drive systems lacking the controllable variable output speed characteristics was abandoned early in the study in favor of the variable drive systems since this feature is considered essential for an optimized hoist design.

The basic variable speed systems are usually classed as mechanical, hydraulic, and electrical. In general the simplest and least expensive of these is the mechanical drive system with the electrical and hydraulic system offering higher horsepower capacity and wider speed ranges. Fortunately, the relatively low power requirement of the hoist drive makes the mechanical drive system an excellent candidate for the hoist. Most variable speed mechanical drive systems have a limited variable speed or stepped speed capability and are not classed as having an infinitely variable speed range, such as is possible with a hydrostatic or some electrical drive systems where output speeds can range from zero speed to maximum design speed without sacrificing torque. However, certain mechanical drives do have "infinitely variable speed" within its operating speed range. The dissipative clutch has infinitely variable speeds from 100 percent down to about 10 percent of input speed.

The results of this trade study clearly establishes the dissipative clutch drive as the prime candidate for use in the helicopter hoist drive.

Table 6-2

FLYWHEEL/HOIST DRIVE

Type	Configuration	Controllability Infinite Variable	Eff. %	State of Art	Weight Lb (1)
Elect.		Yes	70-75	Yes	52-Motor 52-Gen.
Hyd. Trans. without Clutch		Yes	80-85	Yes	72
Hyd. Trans. with Clutch		Yes	97	Yes	100
Dissipative Clutch		Yes	10-98 (2)	Yes	40
Mech. Trans. Fixed Ratio		No	85-95	Yes	35
Mech. Trans. Variable Ratio		Yes	75-90	Yes	120
Hydro Kinetic Constant Fill		No	98 (4)	Yes	Heavy
Hydro Kinetic Variable Fill		Yes	0-98	Yes	Heavy

Notes: (1) Weight of commercially available unit
(2) Wide range due to slip range assumed 10/1
(3) Generally classified as friction or traction drive
(4) Full rated speed only

CABLE HANDLING TECHNIQUES

The method used for cable handling on the present hoist is the capstan type drive system. This consists of routing the cable over two parallel driving drums and through a level wind mechanism into a storage drum under low stress and constant tension.

A second method used for handling cable is the powered drum drive system. This consists of driving the cable storage drum directly with the cable under load tension. Normally, some type of level wind mechanism is used in conjunction with this type of drive, either an independent level wind or a system which moves the power drum longitudinally.

Level wind mechanisms are used primarily to insure consistent cable wind on drum by controlling fleet angle. This is considered necessary because of the characteristics of the wire cables normally used on hoists. The cable must be wound uniformly on the drum with no space between adjacent strands and no bunching on any part of the drum. Otherwise, the cable tends to foul, backwind, or jump as it comes off the drum.

A series of tests were run to evaluate the "self-level wind" characteristics of the normally used wire cable, the swaged wire cable, and the plastic braided rope. The latter two are being considered as alternates to the normally used cable for this hoist application. (Reference Cable Test Report, Appendix B.)

The results of the tests indicate the plastic braid rope has the best level wind characteristics at the larger fleet angles; the swaged wire rope was slightly worse; and the normally used wire cable was worst of all.

One of the design objectives is to retrofit the present hoist configuration which requires a short distance from the drum to the power sheave; the combination of fairly wide drum width and short center distance results in large fleet angles which the plastic rope tolerated better.

Thus, this self-level wind ability makes it possible to develop a hoist with a powered drum and without level wind mechanism which will result in a more compact, lighter unit which will fit within the space constraints of the present system.

The compliance of the plastic braid is an advantage for self-wind design since even with the poor level winds experienced with large fleet angles the plastic rope came off the drum smoothly with no variation in wind-off speed. This compliance is also helpful in reducing the acceleration and deceleration loads experienced near the top and the bottom of the lift cycle.

Section 7
PRELIMINARY HOIST DESIGN

Figure 7-1 shows a preliminary layout of the hoist design assembly which incorporates the best overall configuration based on design studies, trade studies, and tests conducted during this program. The major components of the design include the:

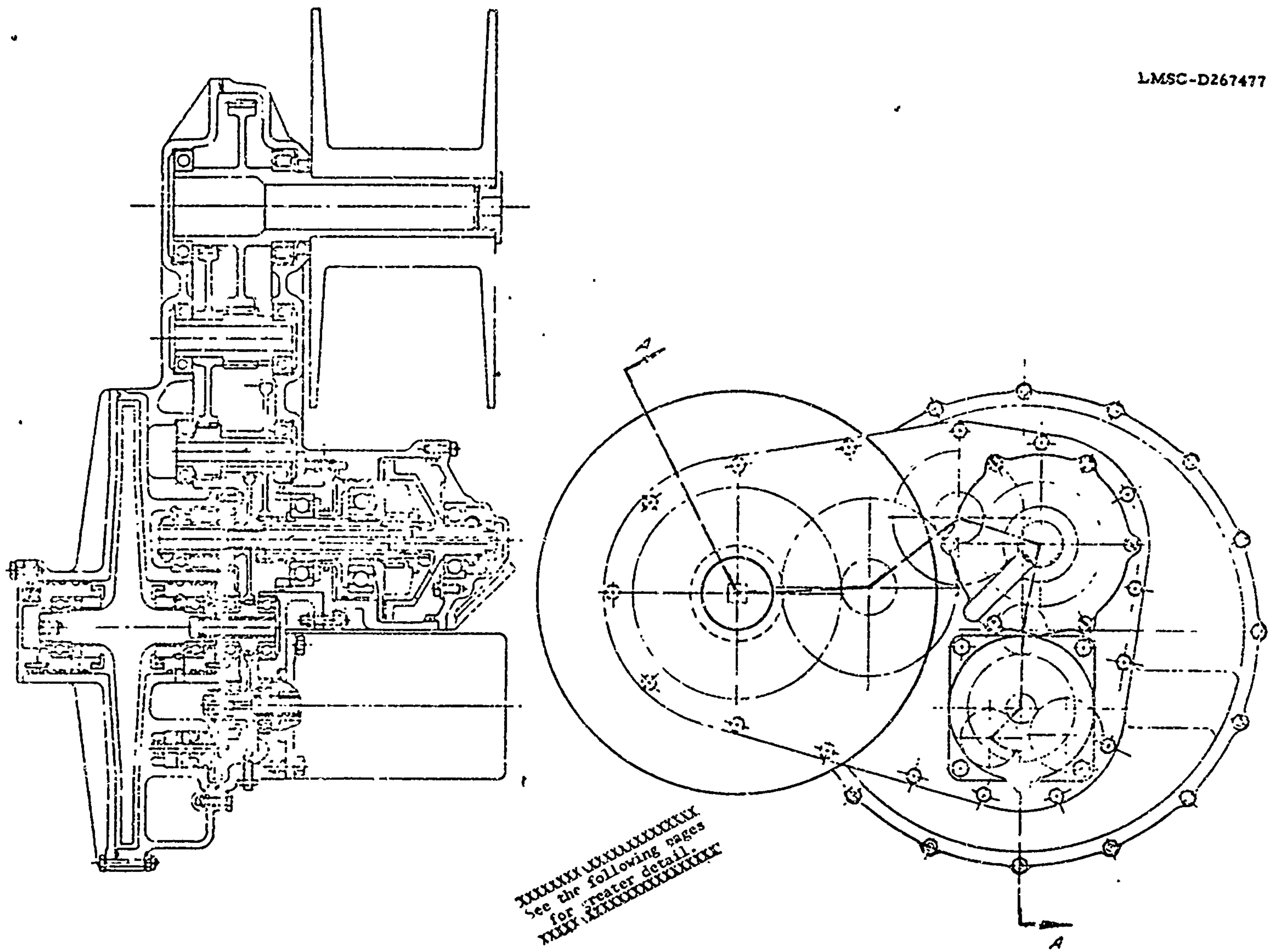
- Flywheel drive motor
- Flywheel
- Drive system clutch and brake
- Drive gears and shafts
- Housing
- Rope drum

FLYWHEEL DRIVE MOTOR

The spin-up of the flywheel to be used for energy storage for the high performance helicopter hoist could be accomplished by a hydraulic, pneumatic, or electric motor. However, since neither hydraulic power or compressed air is available in suitable form, the most obvious choice of motor type is electric. In this way no incompatibility with the existing hoist-to-helicopter interface results.

Electric Interface

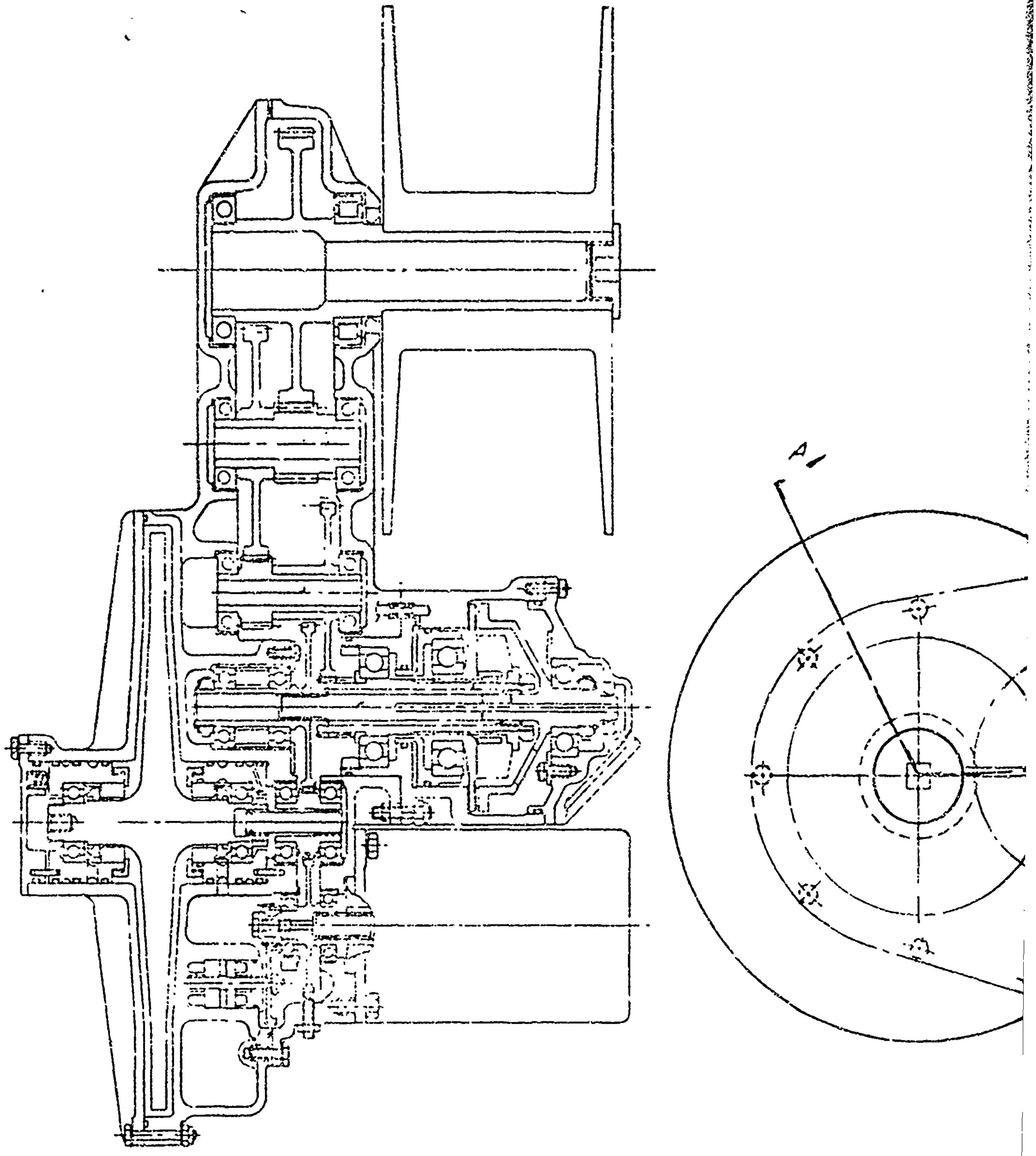
The presently operational hoist uses a maximum armature current of 110 amperes and a field current of 6.5 amperes at a nominal 28 V d.c. from the helicopter electrical system. This represents a maximum electric power of 3262 watts. The assumption has been made in this study that the same maximum electric power is available for the new high performance hoist.



SECTION A-A

7.2.a

Fig. 7-1 Preliminary Layout - Hoist Assembly



SECTION A-A

7.2.a

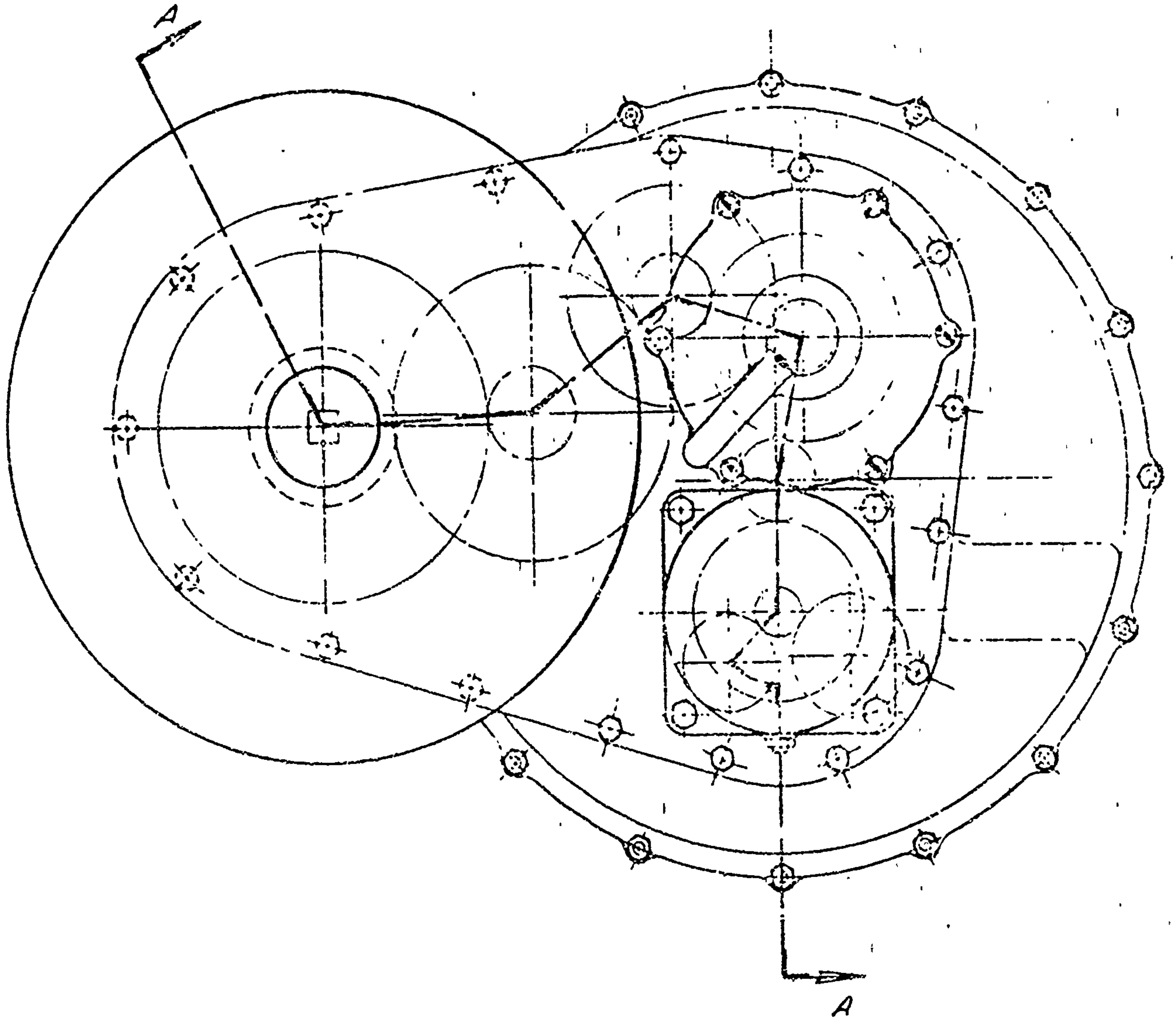


Fig. 7-1 Preliminary Layout – Hoist Assembly

The maximum input motor power then is 4.37 hp, such that even with a 60 percent motor efficiency, 2.6 hp is available for the spin-up of the flywheel.

Electric Motor Speed Range

A preliminary search of suitable d.c. electric motors in the 2-to-2.6-hp range has shown that the maximum normal speed range for 28 V d.c. machines of this capacity is from 8,000 to 18,000 rpm. Since electric motors are generally sized by the torque requirement, it is desired to operate the motor at the highest practical speed to minimize motor torque and, therefore, size and weight for the horsepower required. The desirability of matching the motor maximum speed to that of the flywheel (operational speed range from 14,000 to 28,000 rpm) in order to eliminate the need for a speed increaser between motor and flywheel is also obvious.

A detailed survey was then made of all known suppliers of high performance d.c. motors of the type required for the new hoist. The vendors contacted in this survey are listed below:

Airborne Accessories Corp., Hillside, N. J.
Ajax Corp., Rochester, N. Y.
Ametek/Lamb Electric, Kent, Ohio
B&B Motors & Controls Corp., New York, N. Y.
Baldor Electric Co., Fort Smith, Ark.
Bogue Electric Mfg. Co., Paterson, N. J.
Borg-Warner Corp., Morse Chain Div., Ithaca, N. Y.
Borg-Warner Corp., Pesco Products Div., Bedford, Ohio
Brook Motor Corp., Chicago, Ill.
Century Electric Co., St. Louis, Mo.
Demag Material Handling Corp., Solon, Ohio
Doerr Electric Corp., Cedarburg, Wisc.
Eaton, Yale & Towne Inc., Dynamatic Div., Kenosha, Wisc.
Electronic Specialty Co., Thomaston, Conn.
Galis Mfg. Co., Fairmont, W. Va.
Gerbing Mfg. Corp., Elgin, Ill.
Hitachi America Inc., New York, N. Y.
G. K. Heller Corp., Las Vegas, Nevada
Hoover Electric Co., Los Angeles, Ca.
Marathon Electric, Wausau, Wisc.
North American Rockwell, Boston Gear Div., Quincy, Mass.

H. K. Porter Co., Inc., Pittsburgh, Pa.
Rae Motor Corp., McHenry, Ill.
Reuland Electric Co., City of Industry, Ca.
Robbins & Myers, Inc., Springfield, Ohio
Seco Electronics Corp., Hopkins, Minn.
Singer/General Precision, Inc., Hertner, Cleveland, Ohio
Skurka Engineering Co., Los Angeles, Ca.
Standard Precision Div., Electronic Communications, Witchita, Ka.
Sterling Electric Motors Inc., Los Angeles, Ca.
Task Corp., Anaheim, Ca.
Westinghouse Electric Corp., Aerospace Electrical Div., Lima, Ohio
Westinghouse Electric Corp., Mac Motor/Gearing Div., Buffalo, N. Y.
T. B. Wood's Sons Co., Chambersburg, Pa.

The survey results showed that no standard motors are available in this horsepower range with maximum operating speeds higher than 18,000 rpm. The limit on motor speed is the result of commutation problems.

Two means of achieving the desired 28,000 rpm motor speed were considered. The first alternative is the design of a highly special machine with brush and commutator modifications suitable to permit operation at the desired speed. The design of such motors was proposed by three of the contacted suppliers. The development costs and risks associated with the design of such a new machine appear to be unwarranted in this program. A second alternative for the high speed motor is the design of a special brushless d.c. motor which uses a solid-state inverter to provide a.c. motor excitation such that brushes are eliminated. At present such machines are being manufactured by two suppliers in fractional horsepower sizes. The development of a new brushless d.c. integral horsepower machine for the hoist application is felt to be too expensive at this time although the development risks are small.

As a result of the motor supplier survey, it was concluded that the electric motor for direct drive of the flywheel was not available within the time and cost constraints of the anticipated development program. Thus, it is recommended that a d.c. motor with a maximum speed of 14,000 rpm be used in conjunction with a 2:1 speed increasing gearbox to drive the flywheel. Such a motor is well within the speed range of motors currently being manufactured by several suppliers.

Electric Motor Type

The somewhat unique nature of the load imposed by the flywheel on the electric motor during spin-up and the need to avoid any possibility of overspeed are major considerations in the selection of the motor type to be used.

Series D.C. Motor

Traditionally, drives in which a constant horsepower characteristic is required are accomplished by series type d.c. motors. The diagram showing the basic connection and the torque-speed curve of a typical 2.5 hp series motor is shown in Fig. 7-2. The series motor shown can provide the high torque levels desired for startup and then deliver power to the flywheel at essentially constant horsepower over a wide speed range. However, since the spinning losses (bearing drag, seal drag, and windage) of the flywheel are expected to be quite low even at speeds substantially in excess of the maximum operating speed of 28,000 rpm, the motor torque-speed curve (Fig. 7-2) shows that the series type motor will continue to accelerate the flywheel above the maximum operating speed. Since this possible runaway capability is undesirable, the series motor was rejected for the high performance hoist application.

Shunt D.C. Motor

The shunt field type d.c. motor has a definitely limited no load speed such that overspeed under light load conditions can be avoided. At the same time, the shunt d.c. motor can provide sufficiently high torque for flywheel accelerations and can supply fairly constant horsepower over the flywheel 2:1 operational speed range.

The diagram and torque-speed curves of a suitable shunt type d.c. motor is shown in Fig. 7-3. The effect of variation of the resistance in series with the shunt field is shown by the three torque-speed characteristic curves

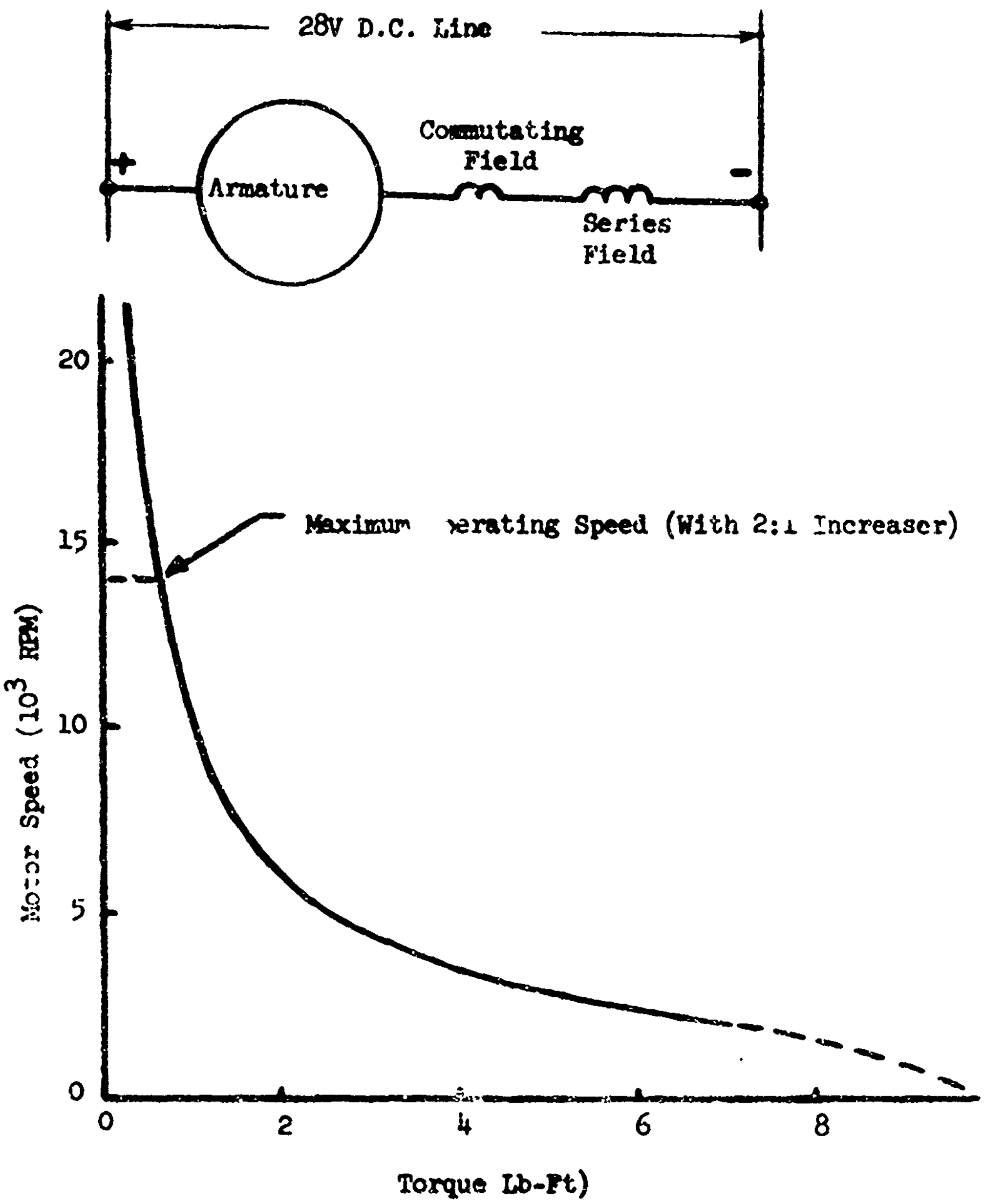


Fig. 7-2 Series D. C. Motor Characteristics

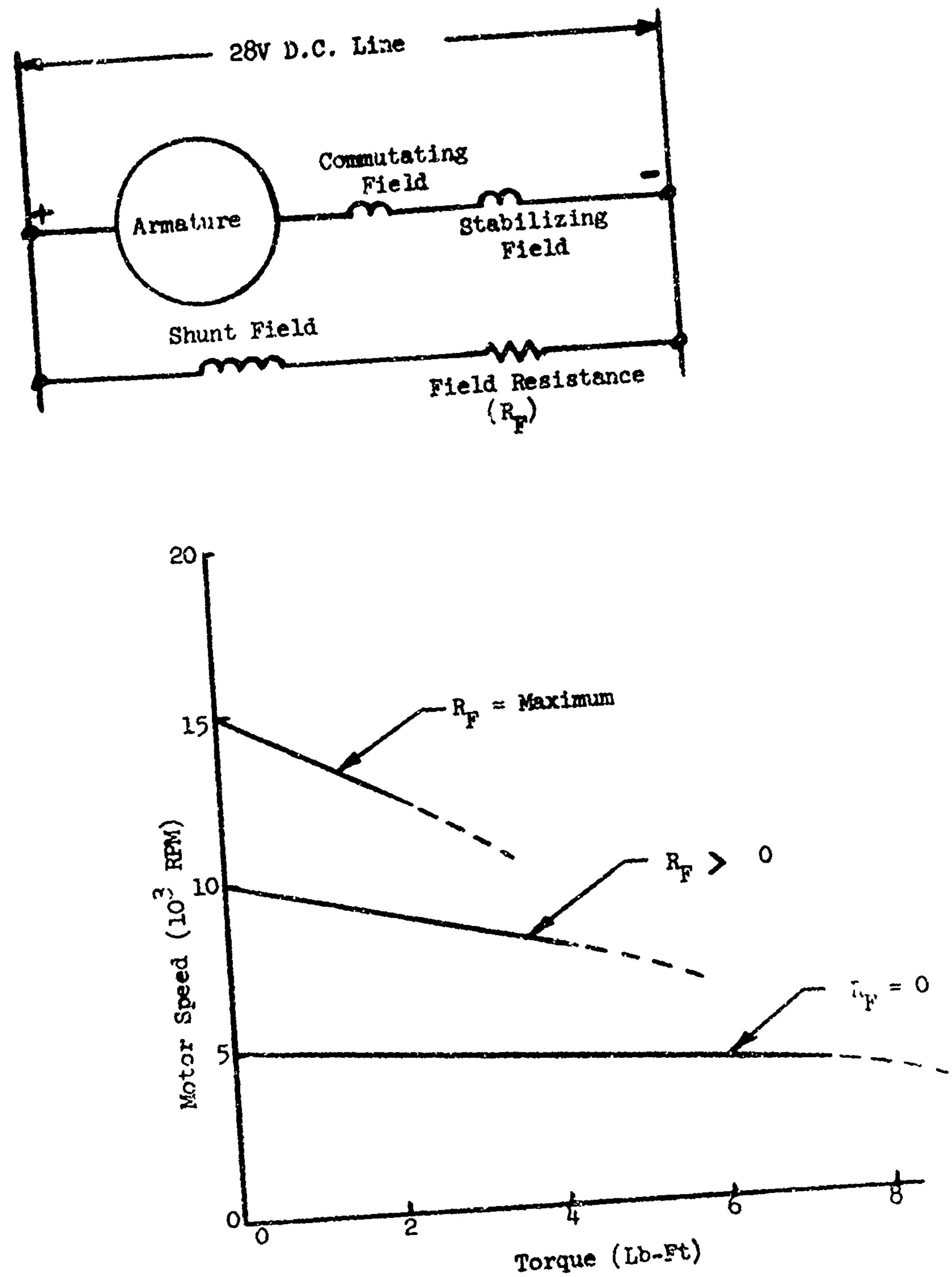


Fig. 7-3 Shunt D. C. Motor Characteristics

of Fig. 7-3. A relatively simple automatic speed control system similar to that shown in the diagram, Fig. 7-4, can be used to provide for spin-up of the flywheel. The typical flywheel acceleration profile is also shown in Fig. 7-4 by the heavy curves superimposed on the shunt d.c. motor characteristics. As the maximum operating speed for the motor (14,000 rpm with a 2:1 speed increaser) is reached, a balance is achieved between flywheel spinning losses and motor torque capability as shown in Fig. 7-4. However, if the flywheel spinning losses for any reason are reduced, the motor speed can only increase to about 110 percent at maximum operating speed. Thus, flywheel runaway is positively avoided by the use of the shunt type d.c. motor. It also should be noted that the typical flywheel acceleration profile follows the desired constant horsepower curve rather closely such that minimal sacrifice in flywheel spin-up time results from the absolute over-speed control provided by the shunt type motor.

The switching of field resistance values as shown in Fig. 7-4 can be accomplished by an automatic relay circuit which uses a speed signal derived from the speed increasing gearbox. As a motor speed of approximately 4,500 rpm is reached, a speed discriminator will operate a relay with make-before-break contacts to add resistance in series with the shunt field of the motor. Similarily, as a motor speed of approximately 9,000 rpm is reached, the speed discriminator will again act to cause the operation of a second relay to further increase the resistance in series with the shunt field. The motor speed control circuit can automatically operate in the reverse manner as a motor speed of about 8,500 rpm is reached as a result of power being taken from the flywheel during hoisting operations.

Finalized Motor Specification

The flywheel drive motor study and vendor survey has resulted in a motor specification suitable for motor procurement in the subsequent phases of the high performance helicopter hoist program.

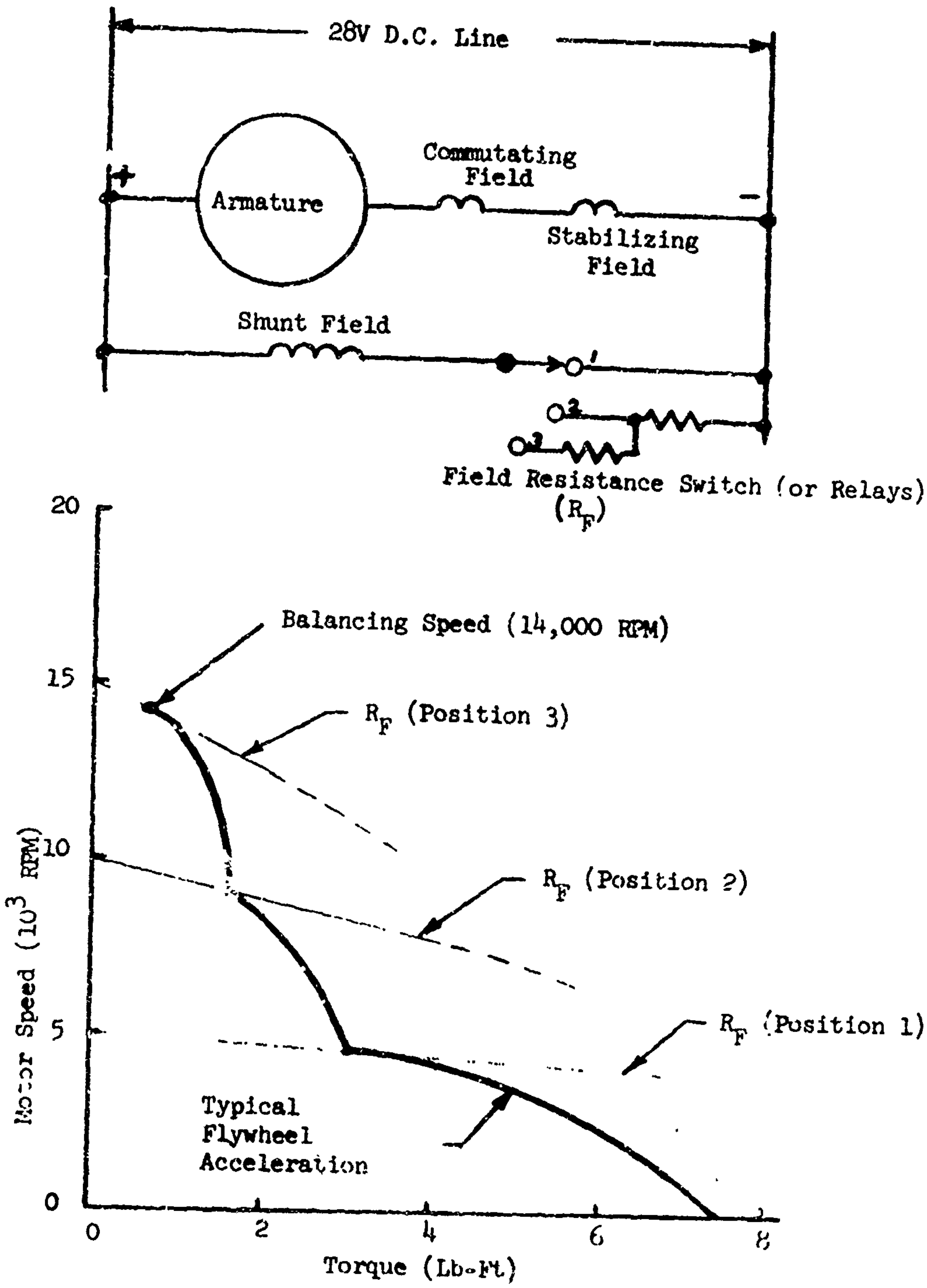


Fig. 7-4 Flywheel Drive Motor Characteristics

The specification for the flywheel drive motor is as follows:

D. C. Voltage: 28 volts nominal
Motor Horsepower: 2.0 continuous
Enclosure Type: Explosion proof-fan cooled
Motor Type: Shunt with commutating and stabilizing fields as required
Direction: Fully reversible
No Load Speed: 16,000 rpm maximum
Full Load Speed: 14,000 rpm
Duty Cycle: 2.6 hp: 15 min. on, 15 min. off, continuous
Efficiency: 65% or higher
Mounting: Square flange
Electrical Connections: All leads brought out 6 in.
Internal Shaft: 0.50 pitch diam., internal spline
Weight: Not to exceed 15 lb

A second round of inquiries to d.c. motor vendors who have responded to the initial survey indicated that a motor in accordance with the above specifications could be supplied by at least four suppliers using conventional manufacturing techniques.

Flywheel Spin-Up Characteristics

The spin-up characteristic for the flywheel of the high performance helicopter hoist is shown in Fig. 7-5. Based on the reasonable assumption that an average of 2.0 hp will be delivered to the flywheel through the 2:1 speed increasing gearbox, the spin-up time for a 73.51 w-hr flywheel is seen to be less than 180 sec. In addition, the plot of Fig. 7-5 shows that the flywheel will be in the usable speed range (14,000 to 28,000 rpm) in approximately 44 sec.

FLYWHEEL

Specific study areas of importance unique to the flywheel such as type of material, flywheel gyrodynamics, and safety analysis are presented in

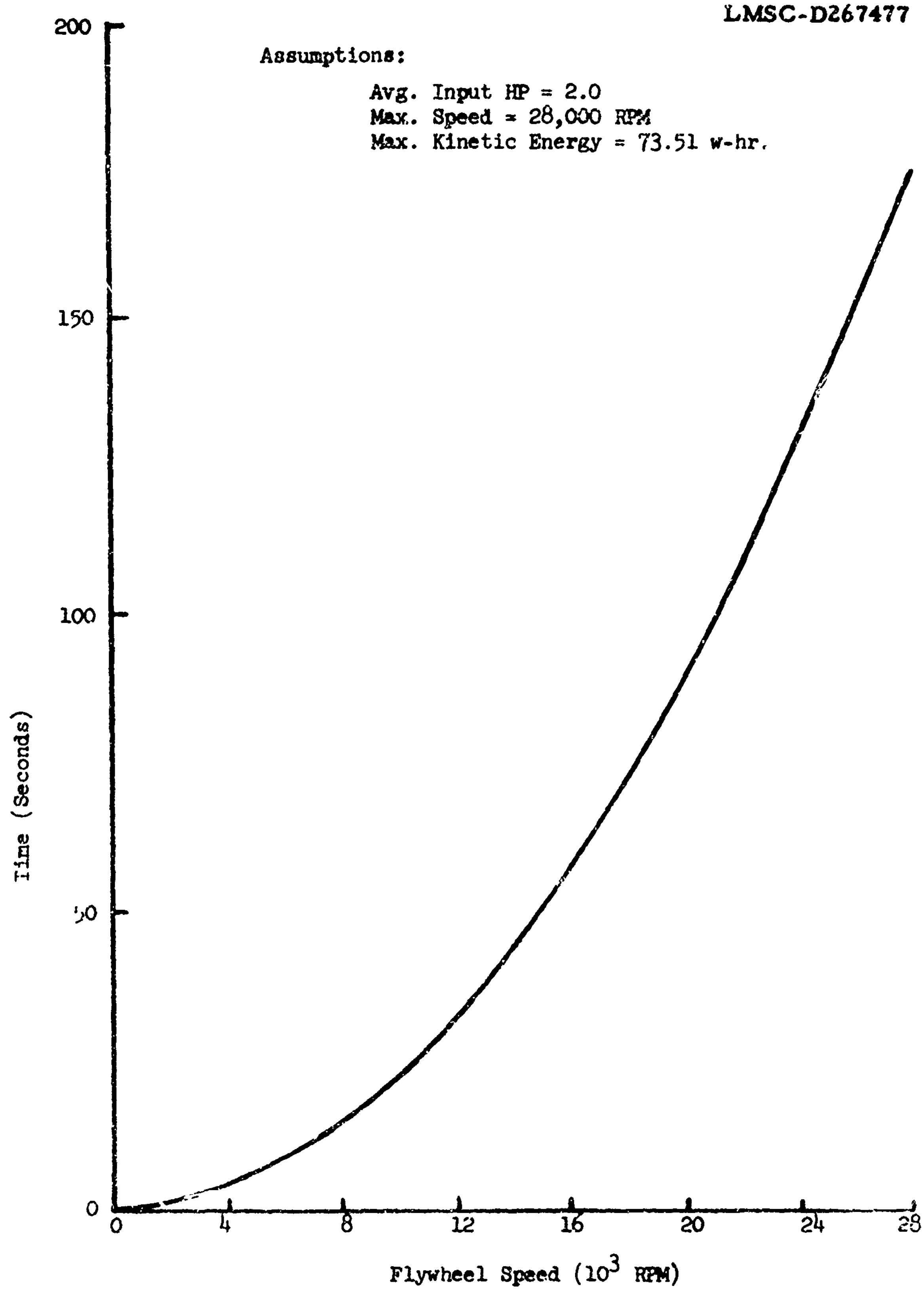


Fig. 7-5 Flywheel Spin-Up Characteristic

Section 4 of this report. Other areas more mechanical in nature are presented her .

Flywheel shaft size is primarily sized by the desire to keep the flywheel operating speed substantially below the "natural frequency" speed, otherwise known as "critical speed." This is done in order to minimize the problems associated with the high speed operation at near or above the critical flywheel speed. Calculated critical speed is 82,000 rpm providing a margin of 3 to 1 over the nominal flywheel speed of 28,000 rpm. Actual margin is less since the calculation assumes infinitely stiff bearings and bearing support structure. Assuming the radial flexibility of the support bearing and support to be 10^6 lb/in. reduces the critical speed to 51,000 rpm, still providing a conservative margin in excess of 1.80.

DRIVE SYSTEM CLUTCH AND BRAKE

Trade studies established the mechanical clutch as the prime candidate for connecting flywheel power to the hoist. Like the flywheel drive motor, it is primarily sized by the torque requirements. Higher operating speeds require less torque and therefore require smaller clutches. The shaft speeds in the hoist range from 28,000 rpm flywheel down to the 240 rpm of the rope drum. Commercially available clutches nominally have maximum operating speed in the range of 3,000 - 8,000 rpm. The desire to minimize weight leads to higher speeds where practicable. Calculations and discussions with clutch vendors established that 12,000 rpm would be compatible with the power requirements of the hoist. Accordingly, a detailed survey was made of known suppliers of clutches and preliminary specifications written for a clutch. Vendors contacted in the survey are listed below:

American Precision Industries Inc., Delaware Div., East Aurora, N.Y.
Borg Warner, Rockford Clutch Division, Rockford, Ill.
Borg Warner Corp., Spring Div., Bellwood, Ill. 60104
Custom Products Corp., Polyclutches Div., North Haven, Conn. 06473
Eaton Towne & Yale, Fawick Div., Cleveland, Ohio 44111
Electrical Company, Div. of Valcor Caye Union, N. J. 07083
Formsprag Clutch Co., Warren, Mi.
Horton Mfg. Co., Inc., Minneapolis, Minn. 55414

General Time, Industrial Controls Div., Farrington, Conn. 06790
Goodyear Tire & Rubber Co., Los Angeles, Ca.
K-M Clutch Co., Ontario, Ca. 91764
Kelsey Hayes Co., Meguon, Wisc. 53092
Machine Components Corp., Plainview, N. Y. 11803
Maxitorq, The Carlyle Johnson Co., Manchester, Conn.
Mercury Clutch, Canton, Ohio
Philadelphia Gear, King of Prussia, Pa. 19406
Pitts Industries, Inc., Dallas, Texas 75234
Precision Specialties, Inc., Pittman, N. J. 08071
The Bendix Corp., Utica, N. Y. 13503
The Conway Clutch Co., Cincinnati, Ohio
The Hilliard Corp., Elmira, N. Y. 14902
The Marquette Metal Products Co., Cleveland, Ohio
Tol-O-Matic, Inc., Minneapolis, Minn. 55415
Twin Disc, Racine, Wisc.
Sawyer Industries, Inc., Arcadia, Ca. 91006
Western Gear Corp., Lynwood, Ca.

Two types of clutches were considered in the survey; a standard clutch for use with a hoist design incorporating selective speed control utilizing two or three fixed gear ratios for varying the lift rate and secondly a controllable variable slip clutch for providing an infinite speed control through its slip range. The basic requirements established during the preliminary design were as follows.

Hoist Clutch and Brake Assembly (Fixed Speed System)

The clutch assembly shall permit the engagement, drive, and disengagement of a powered shaft to a load shaft. This cycle corresponds to the hoist lift cycle where the load starts from rest, is accelerated to hoisting speed during clutch engagement, then runs at full lifting speed until the clutch is disengaged and the load decelerates to rest and is held by a brake. The clutching action during engagement and disengagement shall be smooth and without jerk.

The brake shall be used to hold the load at any position whenever the hoist is not in its lifting mode. This braking action must be coordinated with the clutching action to ensure automatic brake actuation whenever the clutch is

disengaged or whenever the driven shaft is stopped for any reason. In addition, the brake shall be capable of continuously slipping the torque loads to permit the controlled lowering and stopping of the hoist load.

The clutch-brake envelope shape, size, and weight limits have not been established since these parameters are highly dependent upon the type of clutch proposed. LMSC will determine exact requirements after reviewing proposed candidate designs. As a goal, the clutch and brake should be as small and light as possible consistent with good aircraft design practices.

Driving shaft and driven shaft arrangement may be concentric or end to end as dictated by design.

Detail design requirements are as follows:

- Driving side speed, 12,000 rpm
- Driven side speed, 0 - 12,000 rpm
- Driven side load inertia, 0.003 lb in sec^2 (ref. 12,000 rpm speed). Does not include inertia of clutch or brake parts.
- The steady load torques (references to 12,000 rpm) are:
  (1) 16 in.-lb minimum
  (2) 52 in.-lb maximum
- The angular acceleration rate of the clutch driven side shall not exceed 800 radians per sec^2
- The clutch shall be capable of continuously providing a minimum of one smooth engagement-drive disengagement cycles per minute
- Ambient temperature range, $-25^{\circ}F$ to $125^{\circ}F$
- Actuation power source:
  (1) 28 V d.c. electrical, or
  (2) 60 psi lubricating oil pressure
- Life requirement (minimum)
  (1) 12,000 hr of operation without overhaul
  (2) 5,000 clutch engagement-drive-disengagement cycles
  (3) 1,000 brake slipping cycles; maximum brake slip time, 25 sec.

Speed requirement may be reduced if supplier design cannot operate at 12,000 rpm. Reduced speed designs must account for the difference in torques, mass moment of inertia, and acceleration rate which are referenced to 12,000 rpm in these requirements.

Hoist Clutch and Brake Assembly (Variable Speed System)

The requirements for this clutch and brake assembly are identical to the above requirement except the clutch action during the drive portion of the lift cycle must provide controlled torque between driver and driven sides permitting controlled hoisting rates below the clutch "fully-engaged" speed of 12,000 rpm.

The desired variable speed range is 5 to 1 which requires slip speeds ranging from 0 to 9,600 rpm. Duration of slip time to be a maximum of 25 seconds during any lift cycle.

The brake requirements were included with the clutch since its requirements lends itself to a more compact design when designed to utilize common parts with the clutch where possible.

The survey results showed that no standard clutch or brake capable of meeting the requirement exists. However, two companies responded with acceptable special design proposals to meet these requirements. These companies are Philadelphia Gear Co., Synchrodrive Div., and Western Gear Corporation.

The acceptable proposals incorporated slip type clutches which operate by slip shear action of a hydroviscous fluid to transmit torque from the driving surface to the driven surface within the clutch. This "slipping clutch" principle is the same action that is used in the "Synchrodrive" transmissions manufactured by the Philadelphia Gear Corp. On the Synchrodrive units the input shaft is usually at a fixed speed and the output speed is regulated by

controlling the clutch slip torque. The slip torque is regulated by an electro-hydraulic controller which compares the feedback signal obtained from a tachometer or other transducer with a reference signal proportional to the desired output speed. The difference between feedback and reference signal is amplified and operates a servo valve which, in turn, controls the torque of the clutch through a hydraulic piston and actuating mechanisms.

The clutch plate always operates with a film of oil between driver and driven surfaces. The oil is introduced at the inner diameter of the plates where centrifugal force and oil grooving on the plates assures the proper flow and distribution of oil for the lubrication and cooling requirement as well as maintaining the oil film for transmitting torque. Figures 7-6 and 7-7 show typical efficiency and speed modulation characteristics of the slip clutch design.

The clutch torque characteristic as shown in Fig. 7-7 shows the ability of the clutch to maintain a steady, almost flat torque speed curve from about 5 percent to 60 percent of the slip range at full load torque and up to 80 percent at 20 percent of full load torque. This capability to closely modulate the slip speed even at very low load torque is especially important since even light loads will have accurate speed control.

The clutch actuation pressure is provided by a gerator type servo pressure pump mounted to and driven by the same shaft which drives the hoist transmission system lubrication system. All components of the drive system share the same oil supply.

HOIST CONTROL SYSTEM

The control system is an electrohydraulic servo system (Fig. 7-8), designed to provide a variable controlled hoist lifting rate from a crawl speed (approximately 50 fpm) to the maximum 500 fpm. This is accomplished by a hydraulically controlled "slipping clutch" where the input speed/output speed ratio

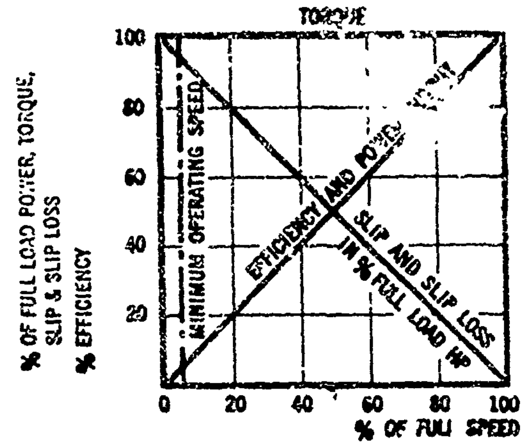


Fig. 7-6 Efficiency and Characteristic Curves, Constant Torque Load (Ref. 12)

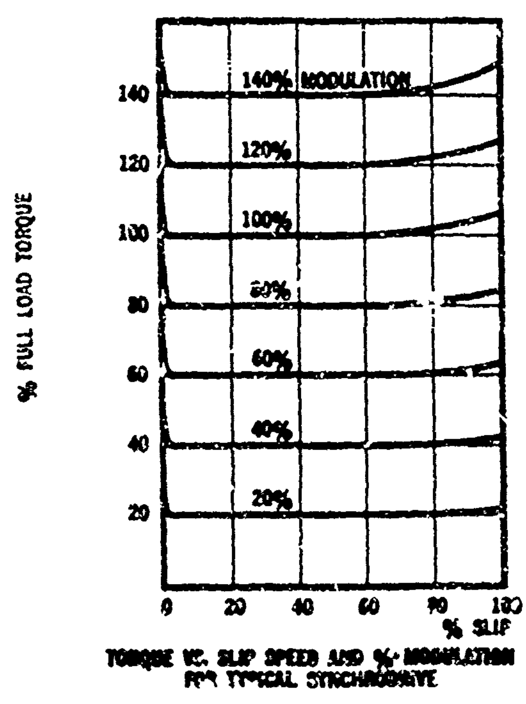


Fig. 7-7 Torque Vs. Slip Speed and Percent Modulation for Typical Scnchrodrive (Ref. 12)

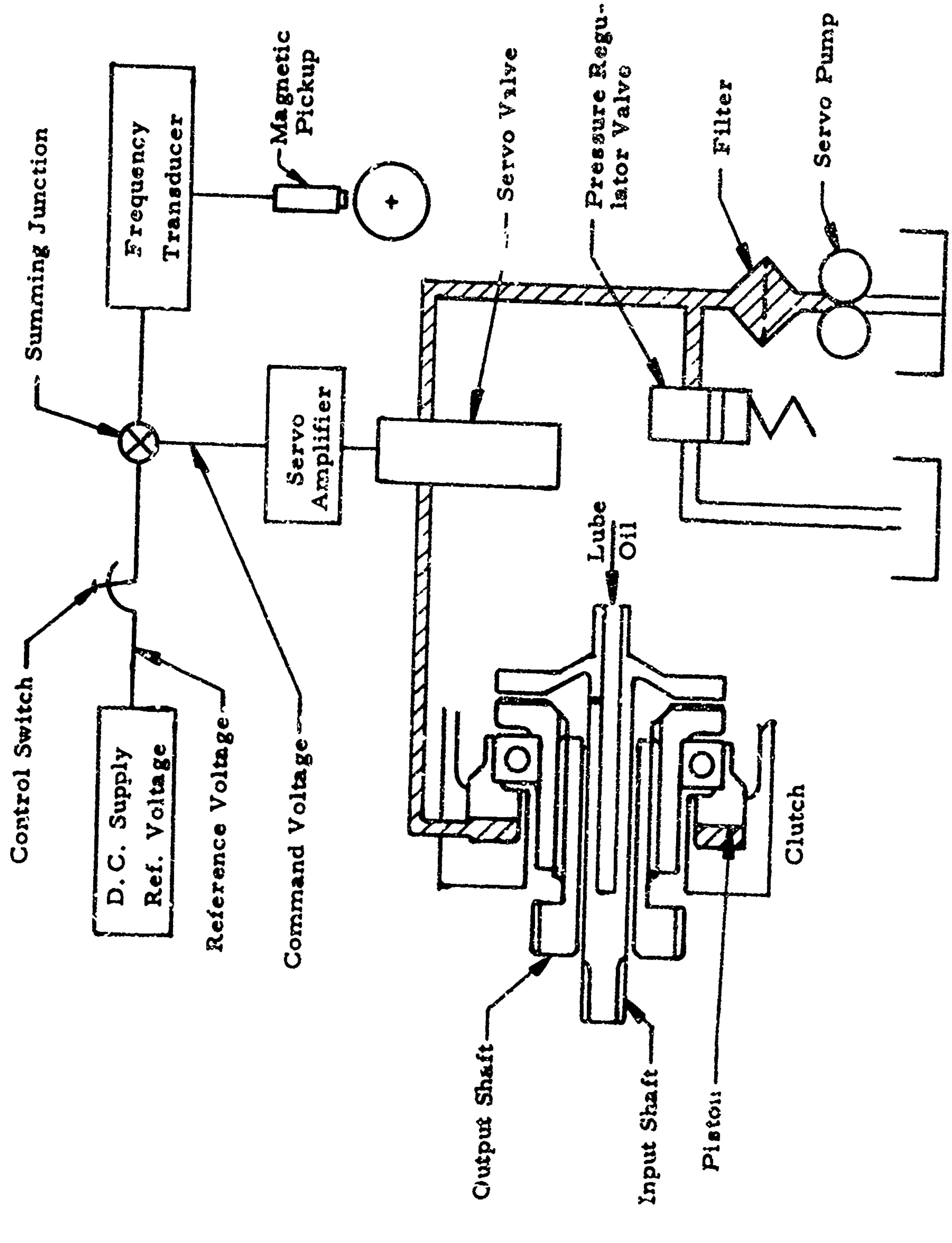


Fig. 7-8 Clutch Control Schematic

is regulated through a speed range of approximately 10 to 1. In operation, the clutch pack transmits torque by hydroviscous shear of the oil film between the driving and driven surfaces of the clutch. The driving plates are mounted on the shaft and the driven plates to the clutch housing. The clutch torque is proportional to the axial clamping force applied onto the clutch pack, thus permitting the relatively simple electrohydraulic method of speed regulation. Increasing the clamp up force increases the clutch torque proportionately decreasing the slip ratio between the input and output speeds. Speed control is obtained by an electronic controller which compares the feedback signal obtained from a magnetic tachometer with a reference signal proportional to the desired output. The difference between feedback and reference signal is amplified and operates the servo valve which, in turn regulates the clutch torque by regulating the hydraulic pressure. The electrical operation of the controller system coupled with the stiffness of the hydraulic system provides response times in milliseconds and speed regulation easily within ±1.0 percent of the maximum speed. Speed regulation as low as 0.25 percent may be achieved if necessary.

Lowering System Control

The lowering control is actuated by the same thumb control used for lift but is otherwise completely independent of the lift control system. The lift modulating clutch is completely disengaged during the lowering cycle, thus removing the flywheel from the hoist drive system. The lowering motor is actuated as the thumb switch is moved in the "down" direction and increases in down speed proportional to the switch motion. The spring loaded switch is designed to provide the operator with a speed versus thumb pressure relationship. The spring also provides a "dead man" emergency system stop action by automatically returning the switch to the neutral off position whenever the operator's thumb is removed from the switch.

FLYWHEEL ANCILLARY REVIEW

The success of the flywheel installation depends largely upon the power function of mechanical ancillary components such as the flywheel support bearings, flywheel housing seals, and vacuum pump. LMSC has conducted within the past several years a number of studies and test programs related to these items. The experience gained has heavily influenced the selection and design of these components discussed in the following paragraphs.

Bearings

Findings from previous LMSC studies (4) and analysis of specific requirements of the rescue hoist flywheel resulted in the selection of simple conrad type bearings with cage design and dimensional tolerances suited to high speed running characteristics.

The basic bearing parameters are:

- Bearing Size

Bore diameter	20 mm
Outer diameter	42 mm
Width	12 mm

- Dynamic Drag Torque: 0.28 in.-lb, max. at 28,000 rpm
- Load and Speed Schedule - per table in Appendix A2
- L_{10} life - 7,000 hr minimum per schedule

The flywheel is supported in a straddle mount arrangement between a pair of single-row, deep-groove, conrad type ball bearings. The bearing mounting arrangement "fixes" the inner race of both bearings to the flywheel shaft. The outer race of only one bearing is fixed while the other is spring loaded axially to effect a face-to-face duplex mounting arrangement. This arrangement locates the flywheel shaft while maintaining sufficient preload on the bearings so that the balls are in contact with the raceways at all times, thus eliminating the possibility of ball skidding. Previous LMSC studies

show that a lower cost special design with nominal ABEC 1 tolerances except for specified concentricity and roundness limits would work satisfactorily for production design quantities. The roundness, face-to-race squareness, and conforming raceway concentricity would provide the necessary high speed running characteristics comparable to an ABEC 5 bearing. However, prototvpe hoists will utilize bearings made to ABEC 7 tolerances, normally stocked by manufacturers for the machine tool industry. The ball retainer in either case is a lightweight, outer-race riding type suited to the high speeds. Lubrication and cooling oil for these bearings are provided by two oil jets per bearing impinging into the space between the ball retainer and inner race. One oil jet per bearing is normally sufficient but two per bearing is used to provide redundancy as insurance against oil starvation in case of a clogged oil jet. The oil jet orifice is 0.030 in. in diameter as a result of experience which establishes this as the smallest practical jet diameter normally able to pass foreign materials, e.g., lint, metal particles, and products of wear commonly found in transmission oil systems.

The violent oil misting action created when the oil jet impinges the rotating bearing race and balls also supplies lubrication for the rotary face seals. Large drain passages are provided to ensure thorough draining and scavenging of the "used oil" from the space surrounding the bearings and seals. This will ensure against unwanted heat and absorption of power due to churning pockets of oil.

The bearing bore diameter selected is based principally on the minimum diameter shaft that can be used and still maintain the critical speed of the flywheel above its rotational speed. Operation above critical speed changes the spin axis from the bearing axis to the center of gravity of the flywheel. For example, a flywheel with an imbalance of 0.001-in. displacement, will have its center of mass 0.001 in. from the bearing axis. At speeds below critical, the flywheel center of mass will orbit around the bearing spin axis causing a rotating centrifugal force on the bearings.

The formula for centrifugal force is:

$$CF = mr\omega^2$$

Where:

CF = centrifugal force (lb)

m = mass (lb-sec^2/ft)

r = displacement of CG from spin axis (ft)

ω = rotational speed (rad/sec)

Figure 7-9 illustrates the imbalance plotted against rotational speeds generally applied to rotating components common to industry. An imbalance of 0.0002-in. displacement for the flywheel assembly is used as the practical limit for simple balancing equipment. The centrifugal force created by this imbalance is the steady rotating force imposed upon the bearings.

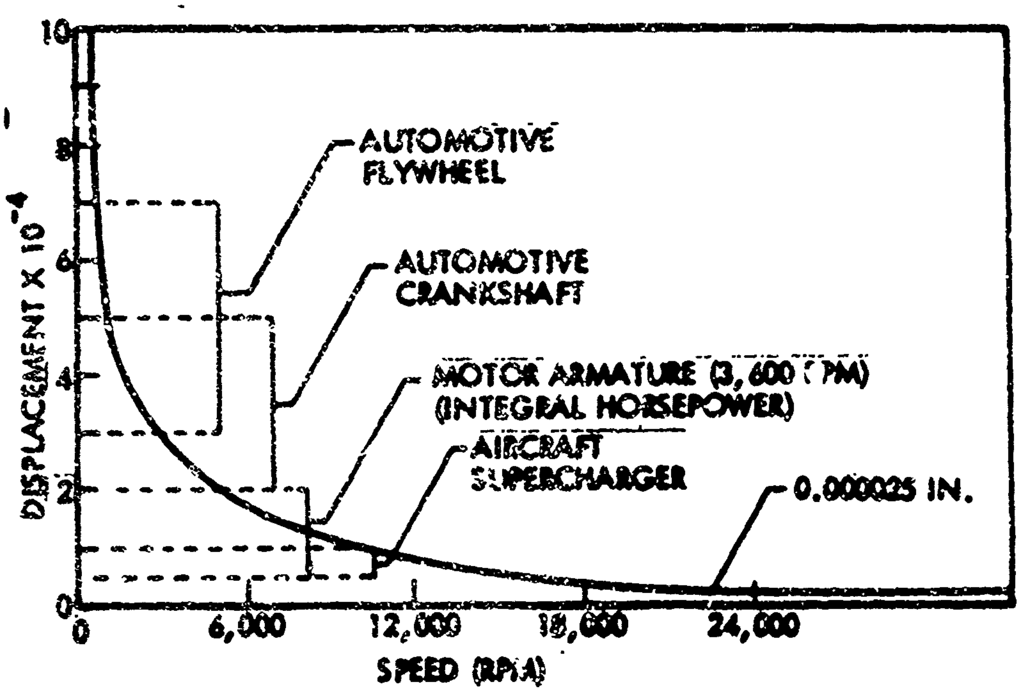


Fig. 7-9 Flywheel Imbalance Displacement Vs. Rotational Speed

Vacuum Pump

In order for the flywheel assembly to function efficiently, it must be housed in a chamber permitting it to rotate in a very low gas pressure environment. Windage losses with even small amounts of air surrounding the flywheel can be detrimental by incurring a heavy horsepower loss to the system. The hoist flywheel is designed to operate in a vacuum level under 5.0 mm of mercury. This pressure level was determined primarily by the two power absorbing functions shown in Fig. 7-10. These are: (1) windage loss, and (2) vacuum pump power requirement. The curves representing flywheel windage loss and vacuum pump power requirement cross at about 4.0 mm Hg indicating the sum total of these two losses to be the lowest at that point.

The vacuum level can be classed as follows:

• Low vacuum	760 to 25 mm Hg
• Medium vacuum	25 to 10^{-3} mm Hg
• High to ultrahigh vacuum	10^{-3} mm Hg and up

Commonly used vacuum systems fall into the low vacuum levels associated with the needs of the processing or materiel-handling industries or into the high to ultrahigh vacuum utilized in the laboratories to simulate space conditions.

The vacuum level required for the flywheel chamber ranges from 30 mm Hg down to under 5 mm Hg, depending on flywheel and flywheel housing design. Industrial pumps are usually of the vane or piston types and the vacuum pressure level is limited to about 28 to 29 in. of Hg with shaft speeds seldom exceeding 3,500 rpm. Figure 7-11 shows the basic types of vacuum pumps that are commercially available. Pumps are arranged from left-to-right in ascending order with respect to vacuum level capability.

The hoist flywheel requires a simple, reliable, maintenance-free pump. Of those depicted in Fig. 7-11, the rotary vane comes closest to these require-

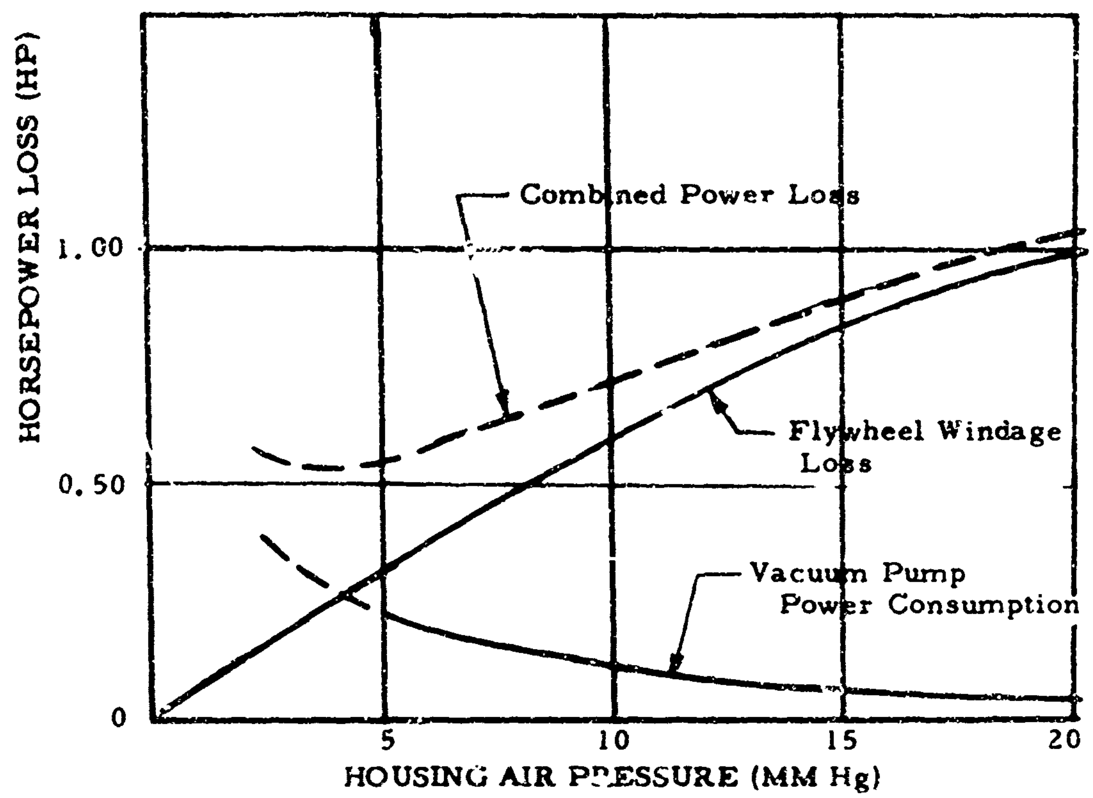


Fig. 7-10 Power Loss Vs. Flywheel Chamber Air Pressure

Curves showing flywheel windage losses vs. flywheel housing air pressure. Also shown is a vacuum pump power vs. air pressure curve and curves combining windage and pump power. Note minimum losses occur at under 5 mm Hg. air pressure.

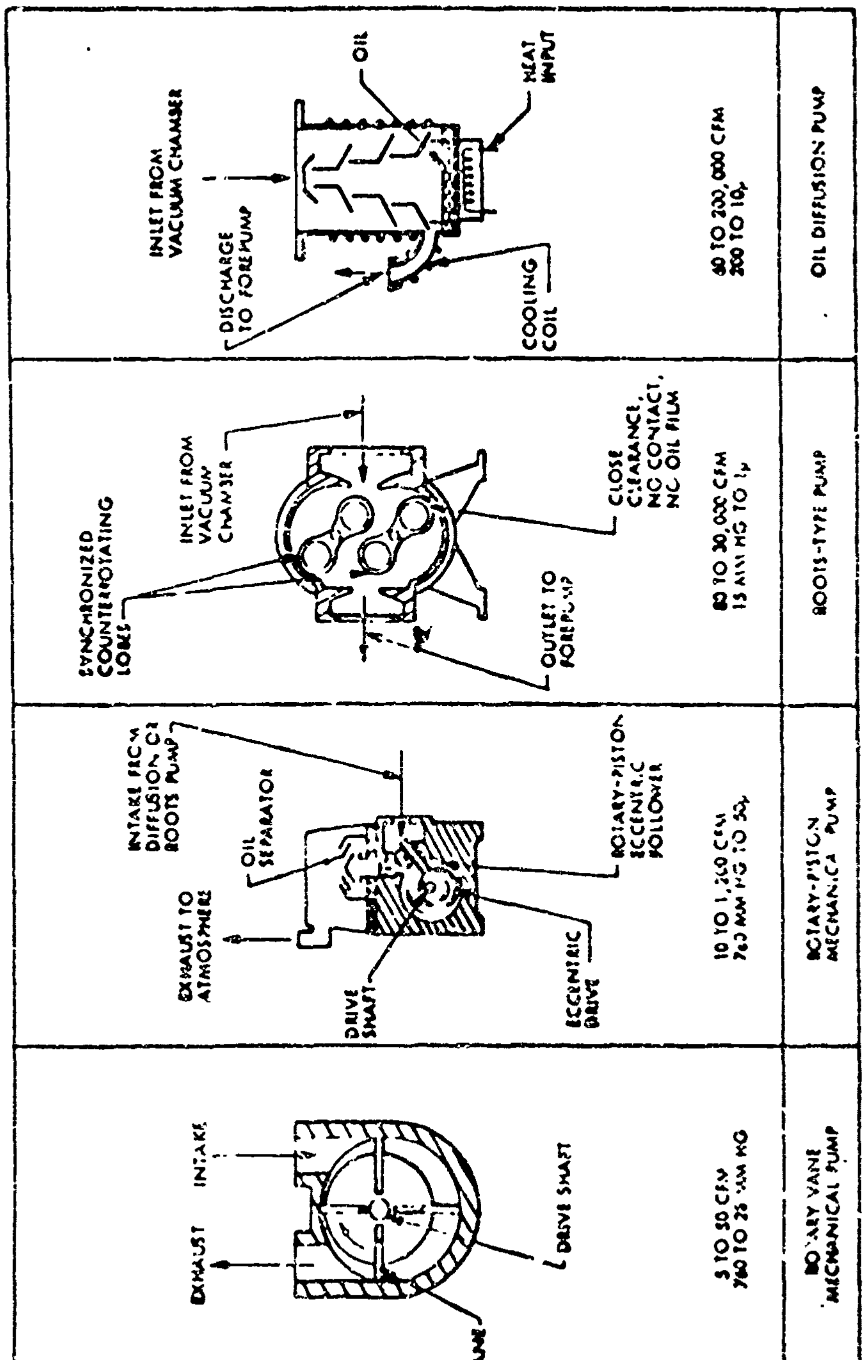


Fig. 7-11 Types of Vacuum Pumps

ments. However, its lowest vacuum levels only approach the upper limits of air pressure that can be tolerated within the flywheel housing. The others have the necessary vacuum capability but are too complex, expensive, and heavy for effective use on the transmission.

The final selection is the gerotor type mechanical pump shown in Fig. 7-12. It is a lightweight, simple, mechanically-driven pump used successfully for many years as a fluid pump for transmissions and hydraulic systems.

The operation of th pump is illustrated in Fig. 7-13. The pumping mechanism consists of two elements, an inner rotor and outer rotor. The inner element always has one less tooth than the outer.

The volume of the "missing tooth" multiplied by the number of driver teeth determines the volume of fluid pumped at each evolution (cubic displacement per revolution). The number of teeth may vary, depending on such design considerations as volume to be pumped, speed, and available pump envelope, but the inner element always has one less tooth than the outer.

As the toothed elements, mounted on fixed centers but eccentric to each other, turn, the chamber between the teeth of the inner and outer elements gradually increases in size through approximately 180 deg of each revolution until it reaches its maximum size – equivalent to the full volume of the "missing tooth." During this initial half of the cycle, the gradually enlarging chamber is exposed to the suction port creating a partial vacuum into which the liquid flows. During the subsequent 180 deg of the revolution, the chamber gradually decreases in size as the teeth mesh and the fluid is forced out the discharge port.

The pump configuration, consisting of an internal gear and mating rotor, provides inherent advantages suited to the higher speeds associated with the flywheel transmission. Both elements revolve in the same direction and the relative speed between them is proportional to the tooth ratio; thus,

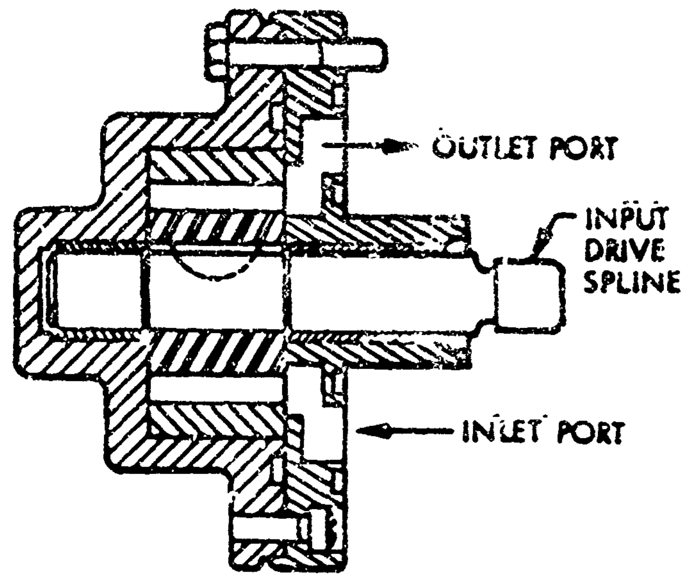


Fig. 7-12 Gerotor Type Mechanical Pump – Cross Section

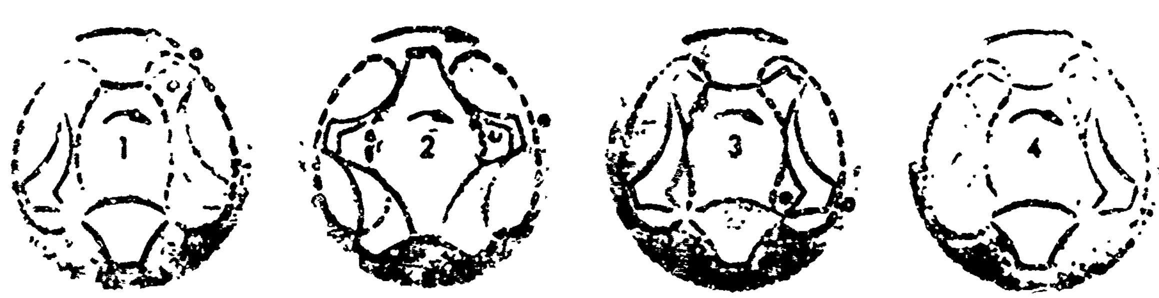


Fig. 7-13 Operating Cycle of Gerotor Pump

high shaft speeds result in low relative pump element speeds. Rotor speeds of 7,000 to 8,000 rpm on medium-sized pumps (2-in. diam.) are common, and speeds approaching 60,000 rpm have been run successfully on smaller units.

The basic sizing formula for a vacuum pump is:

$$C_p = \frac{2.3}{\Delta T} (V) \log \left(\frac{P_1}{P_2} \right) + \frac{Q_o + Q_2}{P_2}$$

where

C_p = pump load capacity (cfm)
ΔT = pumpdown time (min)
V = free volume in flywheel housing (ft^3)
P_1 = initial pressure (mm Hg, abs)
P_2 = final or working vacuum pressure (mm Hg, abs)
Q_o = outgassing load (Torr-cfm)
Q_2 = leakage load (Torr-cfm)

The pump size using the formula is 3.20 cfm. Detailed pump calculations are given in Appendix A3.

The first term of the formula is governed primarily by the volume and pump-down time. It represents the pump size required if outgassing and leakage are assumed to be zero. The second part represents the pump size requirement due to outgassing and leakage. The small outgassing surface areas involved and the type of materials used for the flywheel and housing contribute to a negligible outgassing load, conservatively estimated as 0.1 Torr-cfm. The leakage rate Q_L is the most important factor in sizing a pump for a flywheel housing. A plot of pump size versus leakage rate is shown in Fig. 7-14.

Since the primary use of a gerotor type pump is for positive pressure, there was little information available for its use as a vacuum pump. Discussions with engineering personnel representing the W. H. Nichols Company (Waltham, Mass.) provided the necessary impetus for further investigation. Tests were conducted on a two-element pump assembly at the Nichols Company and at the LMSC Ground Vehicle Test Facilities in Sunnyvale, California (ref. Appendix A3). The important parameters under test were as follows:

- Maximum vacuum level attained
- Pumpdown time
- Driving power requirements

The tests simulated the conditions that a vacuum pump would encounter in a flywheel transmission. The conclusion drawn from the test results is that the pump provides the performance requirements necessary to make it acceptable for use as the vacuum pump in the flywheel transmission. Test results are summarized in the following paragraphs.

The pump sustains low air-pressure levels consistently under 5.0 mm Hg, and the pumpdown time to 10 mm Hg never exceeded 25 sec, even at the lowest test run speed of 5,200 rpm. Figure 7-15 shows a typical pumpdown time plot from the test recorder.

It should be noted that the pump used was not designed specifically for the application – it was designed for use as a scavenger pump in a gas turbine. The only adaptation made for vacuum pump operation was to provide approximately a 10-in. head of oil in a standpipe on the discharge port. This ensured continuous lubrication for the rotors and provided "oil sealing" between parts. In normal use of the gerotor-type pump (in a pressure or scavenging application), there is no need for these special features because the fluid flow through the pump is sufficient to lubricate and to disperse heat. However, as a vacuum pump, it does not have the oil flow necessary for lubrication and cooling. A second set of pump elements, adjacent to the

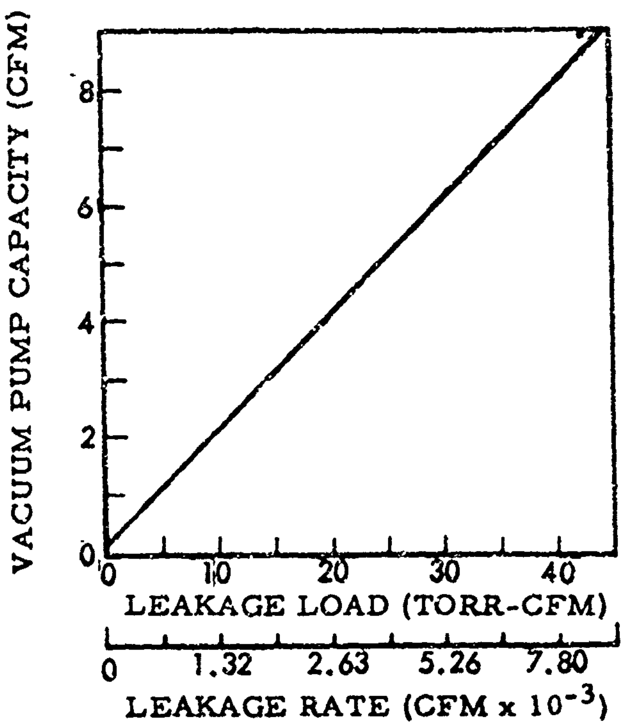


Fig. 7-14 Pump Size Vs. Leakage Rate

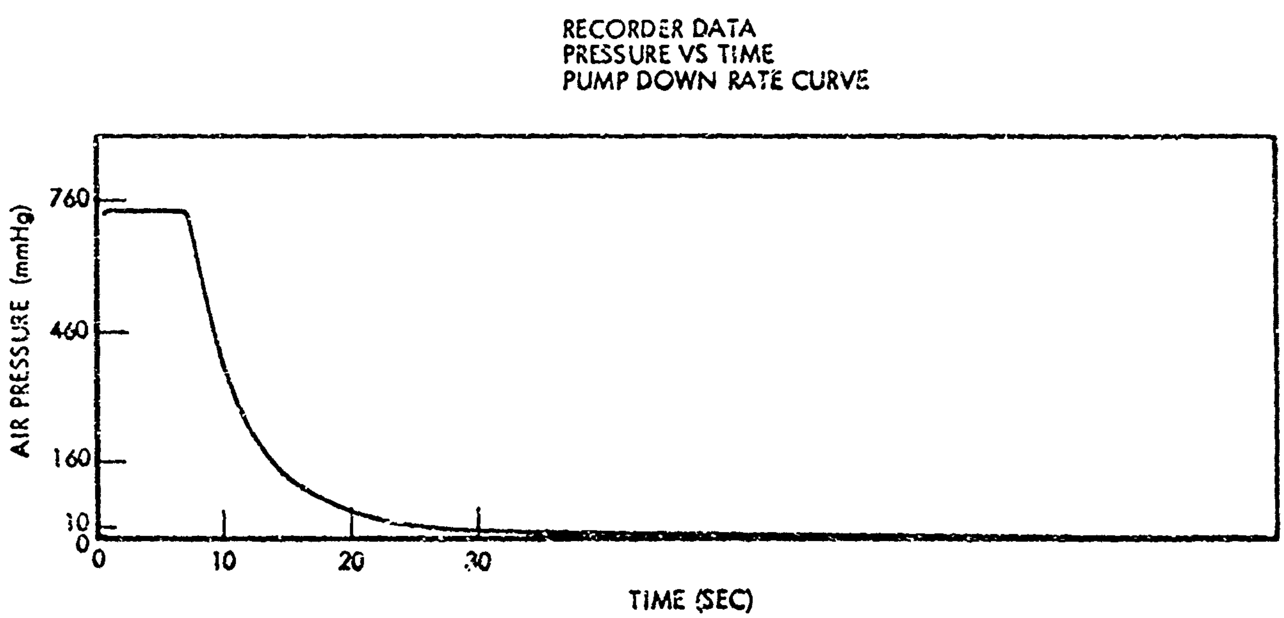


Fig. 7-15 Recorder Data – Pumpdown-Rate Curve

vacuum elements and driven by the same shaft, provide the necessary cooling for the vacuum pump elements and also provides scavenging for the flywheel support bearings and seals.

Rotary Shaft Seals

The rotary seal around the flywheel support shaft must minimize oil and air leakage into the flywheel vacuum chamber to maximize flywheel efficiency and to permit the use of a reasonably small vacuum pump. The leakage rate is the primary factor in determining the pump size.

The importance of low leakage is illustrated in Fig. 7-14 which shows that only a 0.14 cfm pump is required for pumping down the small flywheel housing volume and the added capacity requirement is proportional to air leakage. In other words, the smaller the leak, the smaller the pump.

Labyrinth type seals were eliminated from consideration because of its high leakage rates and only rubbing seals studied.

In the rubbing seal category, a second major breakdown divides the contact type seals into two types. The first is the lip seal with its sealing action provided by an interference fit between a smooth rotating shaft surface and a flexible sealing element. The sealing element is usually made of leather or synthetic elastomers and the interference fit is usually augmented by spring pressure provided by a garter type or finger type spring. The second contact type seal is the "face seal" which creates dynamic sealing in a plane vertical to the shaft axis. This type of seal has two parts – the seal cartridge, consisting of the housing, end face (nose) element, and spring assembly; and the rubbing ring, which is the mating element that provides a smooth flat sealing surface. For high speeds, the end face is normally made of carbon and treated to reduce friction, and the mating ring is made of close-grained cast iron or steel. Table 7-1 is a brief summary of characteristics of the two groups of contact seals.

Table 7-1
SEAL CHARACTERISTICS

Parameter	Lip Seals	Face Seals
Nominal Speed Rating	0 to 3,000 fpm	0 to 50,000 fpm
Life	Good	Good to excellent
Dynamic Friction	Good to poor	Good to excellent
Cost	Low	High
Ease of Replacement	Good	Normal
Misalignment Tolerances	Excellent for axial runout; good for radial runout	Excellent for radial runout; good for axial runout
Fluid Leakage	12 to 48 drops/day	2 to 3 drops/day
Gas Leakage	--	0.1 Torr cfm

In addition to the high speed and low leakage requirement, low drag loss is necessary for maximum system efficiency.

Preliminary figures for power loss were obtained using the coefficient of friction of carbon bearings and applying it to the following formula:

$$hp = \frac{P \cdot \mu \cdot r \cdot N}{63,025}$$

where:

P = axial force (lb)

μ = coefficient of friction

r = mean radius of seal nose (in.)

N = shaft speed (rpm)

Values thus obtained were used for all the preliminary flywheel seal losses.

Additionally, high speed tests were conducted on a Cartriseal face seal. Figure 7-16 shows the drag versus speed relationship of the seal with various nose loads. Seal losses were about double that originally estimated for the test seal but is expected to be reduced down to about the original values by using a dense hard chrome finish on the rubbing ring.

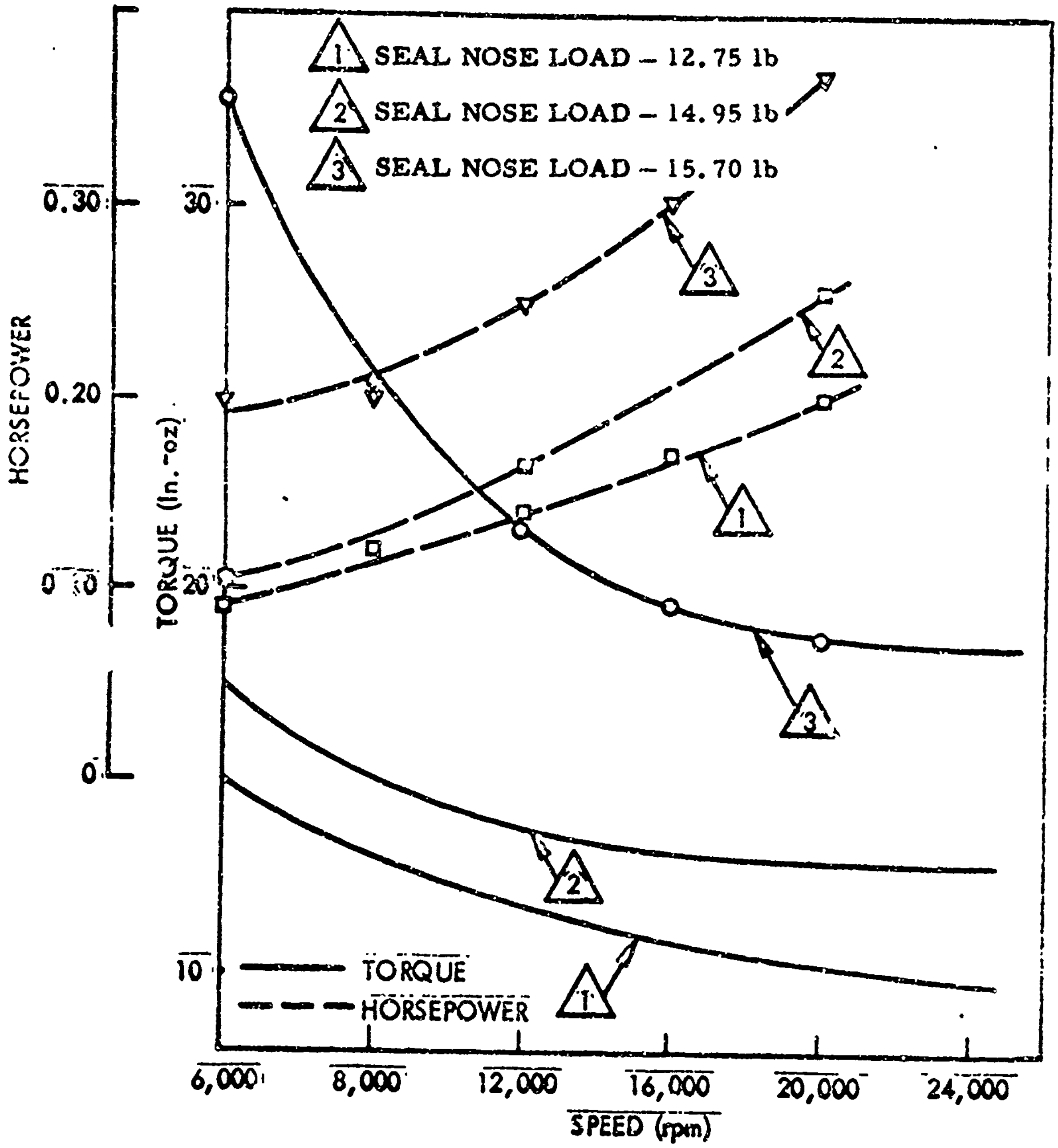
Calculated loss at rated flywheel speed is 0.055 hp per seal.

INSTALLATION CONSIDERATIONS

There are four optional locations in the helicopter for the installation of the hoist and boom assembly. Consultation with military users has indicated the ability to use all locations is not necessary, and the optimum location for the boom assembly is the right forward position. The selection of this location permitted certain design advantages which were used in one of the configurations.

There are certain problems with the present hoist installation which became evident during operation of the hoist in the helicopter cabin mockup (Fig. 7-17). One of the problems is that with the hoist boom arm fully extended the hoist cable passes within 12 in. of the helicopter landing skid; as a rescuee is raised, there is inadequate clearance as the skid is passed (Fig. 7-18). Another problem is that with the jungle penetrator installed, the bottom of the penetrator will not clear the floor with the hook in the up position, so that the boom assembly cannot be rotated inside the helicopter with a load on the penetrator. A third problem is that the hoist portion of the hoist assembly infringes on the needed rescue space which makes it difficult for the hoist operator (Fig. 7-19). There are two installation configurations being considered.

The first configuration (Fig. 7-20) consists of mounting the hoist package near the bottom of a vertical pole in the same manner as the present configuration, with the hoisting rope coming off the drum, going through a sheave



Cartriseal Corp. Seal Part No. 1-1875
(1.795 ID, 2.505 OD, x 0.735 long)

Fig. 7-16 Drag Torque and Horsepower Relationships Vs. Seal Speed

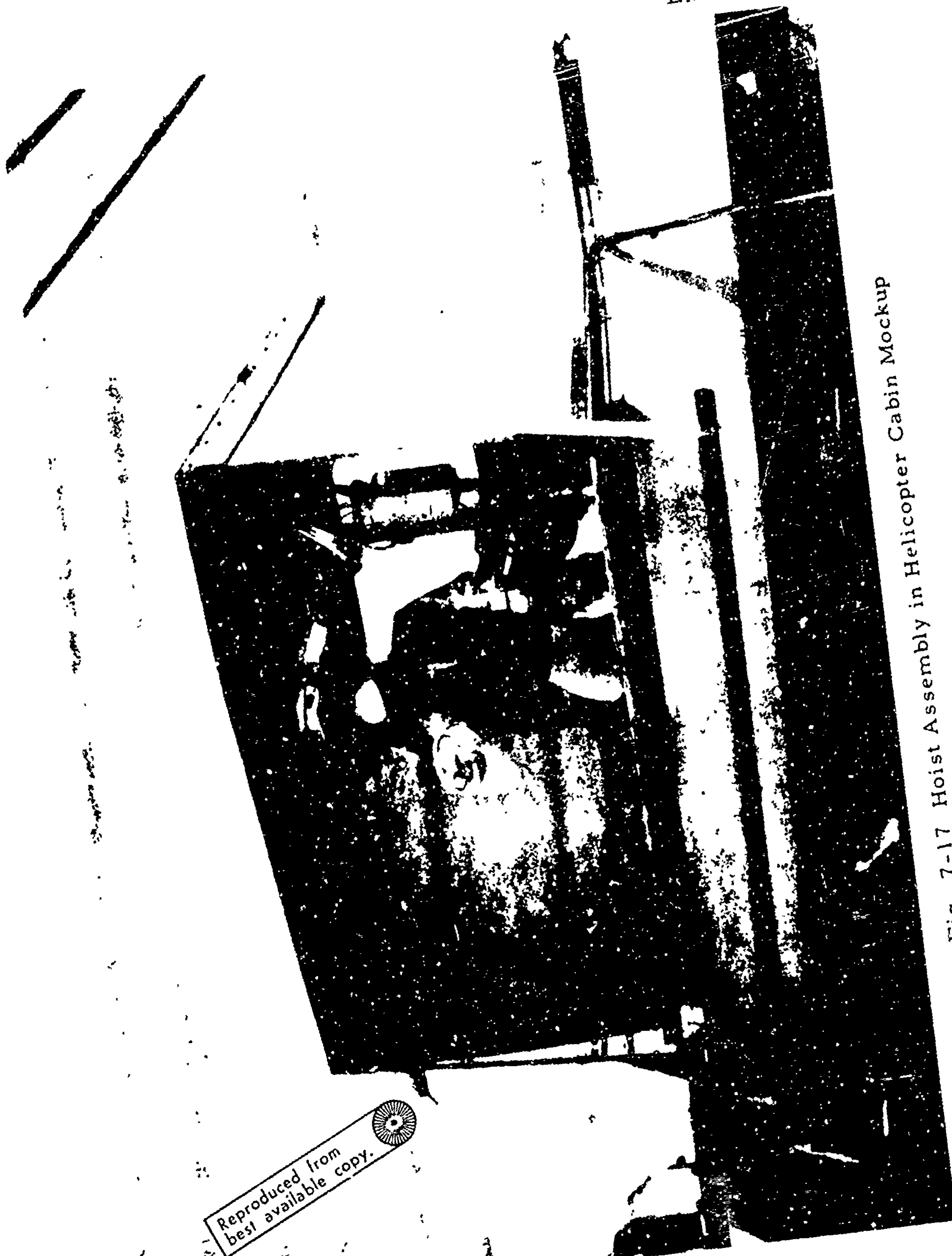


Fig. 7-17 Hoist Assembly in Helicopter Cabin Mockup

Fig. 7-18 Rescuee Passing Helicopter Skid Gear

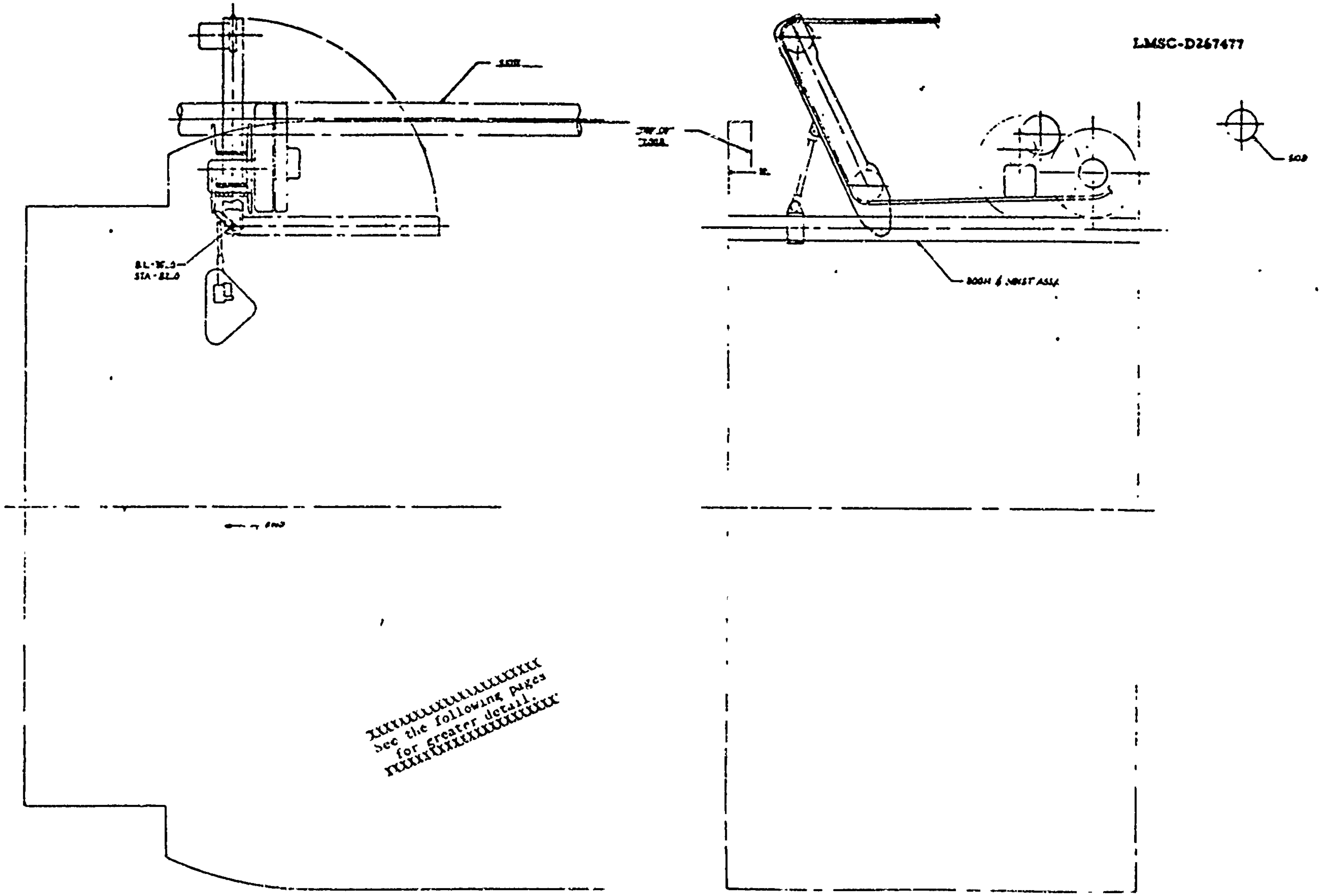
Fig. 7-19 Rescuee Being Swung into Helicopter Cabin Mockup

mounted near the base of a horizontal boom member, then going through a traction sheave mounted in the end of the horizontal boom and on down to the load.

The second configuration consists of mounting the hoist package, either on the ceiling or a vertical pole, placed diametrically opposite the existing hoist installation, taking the hoisting rope across under the ceiling and running along side a vertical pole through a set of roller guides and out of a traction sheave at the end of the horizontal boom member. See Fig. 7-21.

The second configuration has several advantages over the present installation in that by moving the hoist portion to the other side, the lateral moment loading produced by the installation will be minimized. This will allow the use of a longer boom arm without changing the existing moment loading; this also will provide additional clearance (approximately 5 in.) between the hoisted load and the helicopter skid.

Another advantage of this approach is due to the removal of the hoist from the bottom of the vertical pole, thus providing additional clearance in the working area for the operator. This arrangement also lends itself to easy replacement of the hoisting rope during a rescue mission. In the event of a rope hang-up the rope could be cut with the guillotine or a backup cutting device and a replacement rope drum and rope carried on the helicopter could be installed on the hoist, restrung on the boom assembly, and the rescue operation continued without returning to base.



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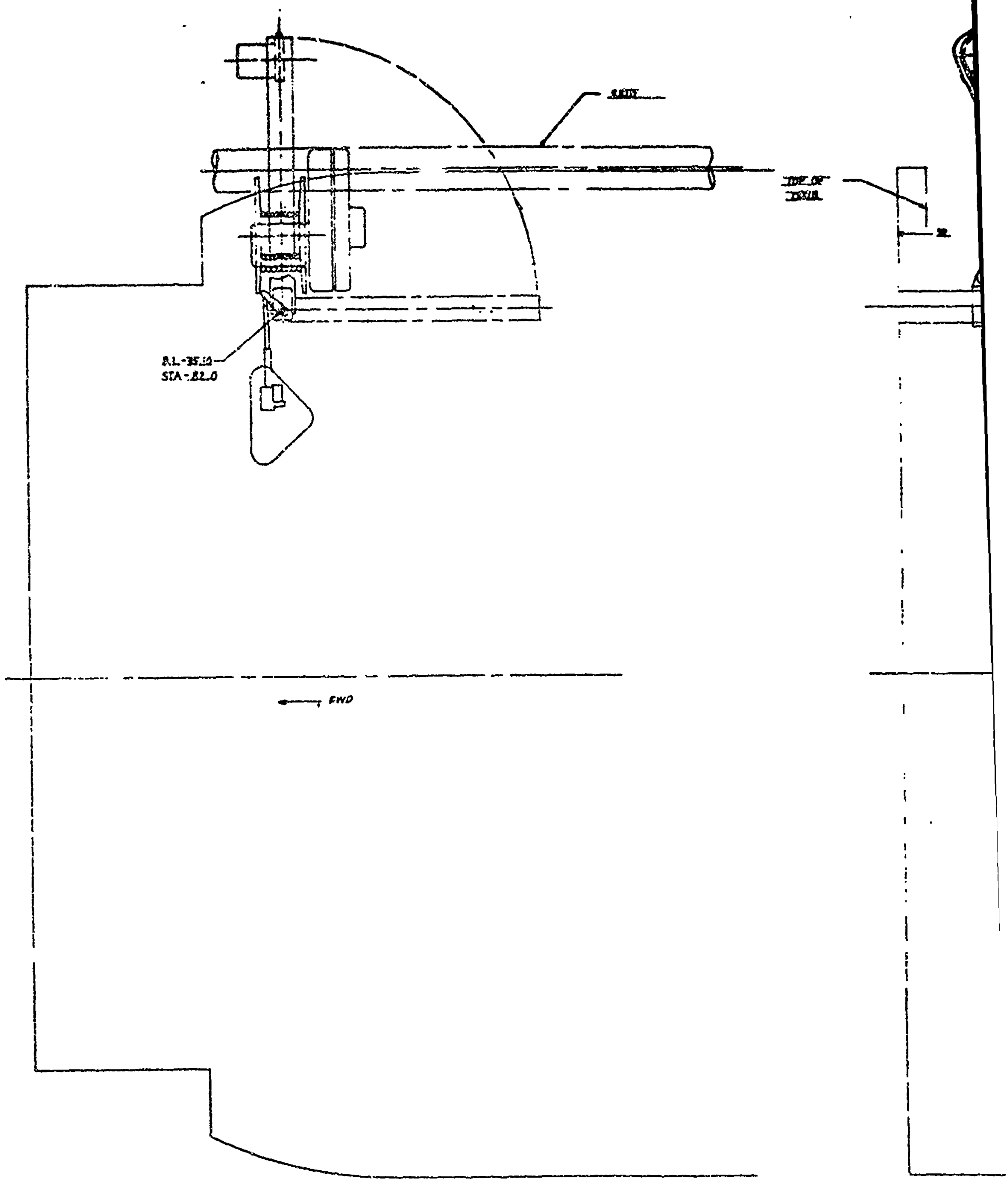
7-39-a

Fig. 7-20 Preliminary Layout - Boom Mounted Hoist



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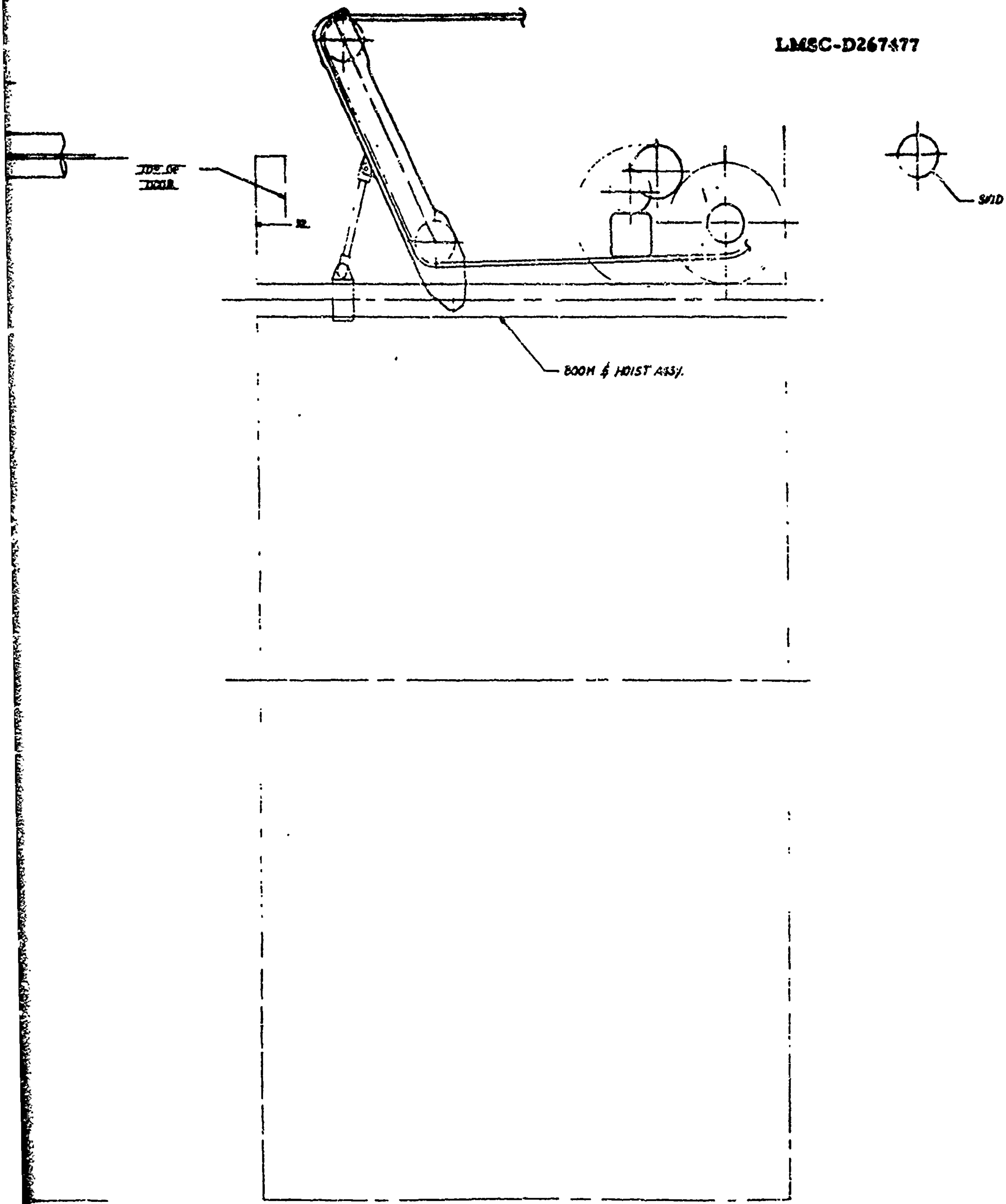

Fig. 7-20 Preliminary Layout - Boom Mounted Hoist

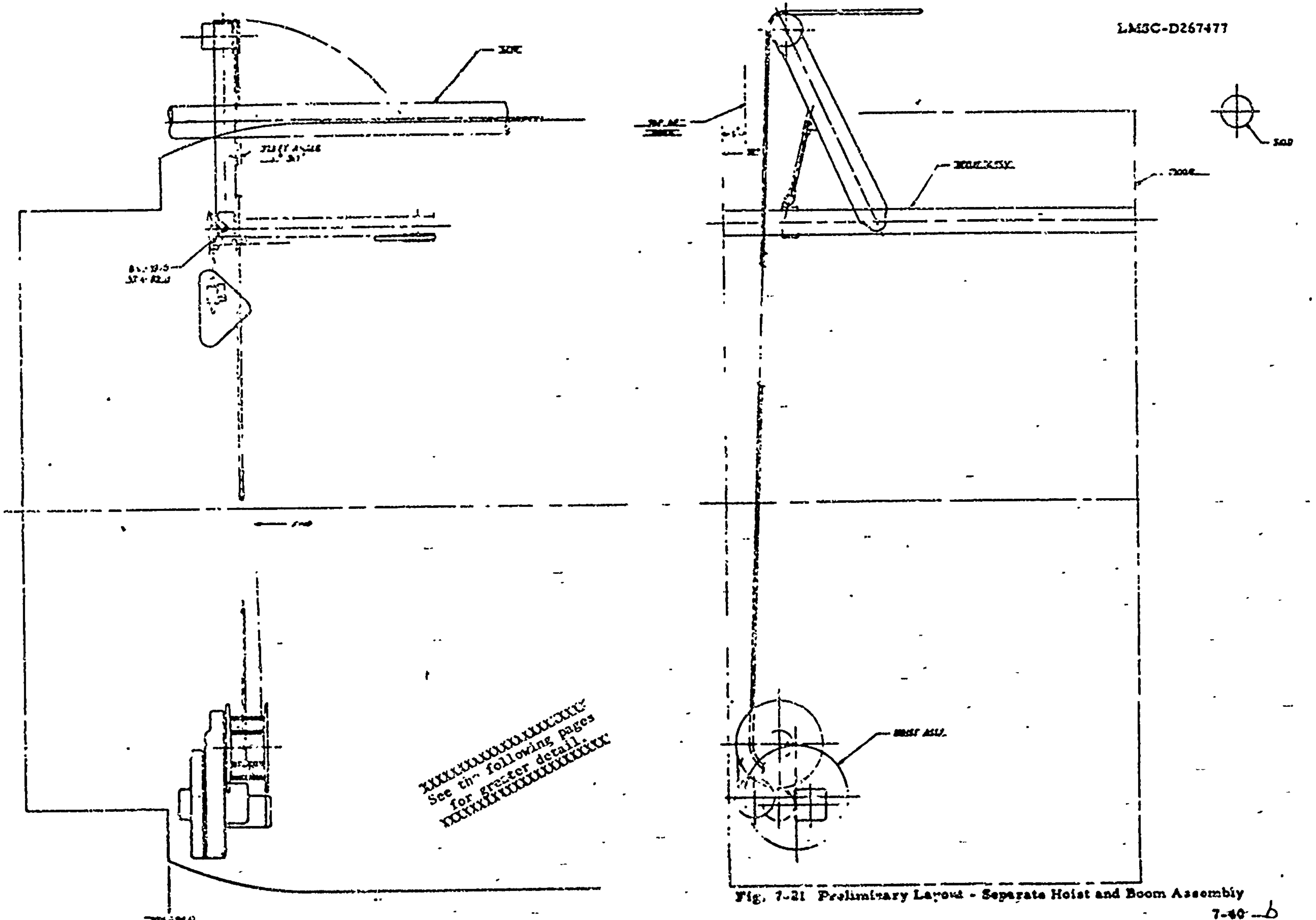

Fig. 7-21 Preliminary Layout - Separate Hoist and Boom Assembly

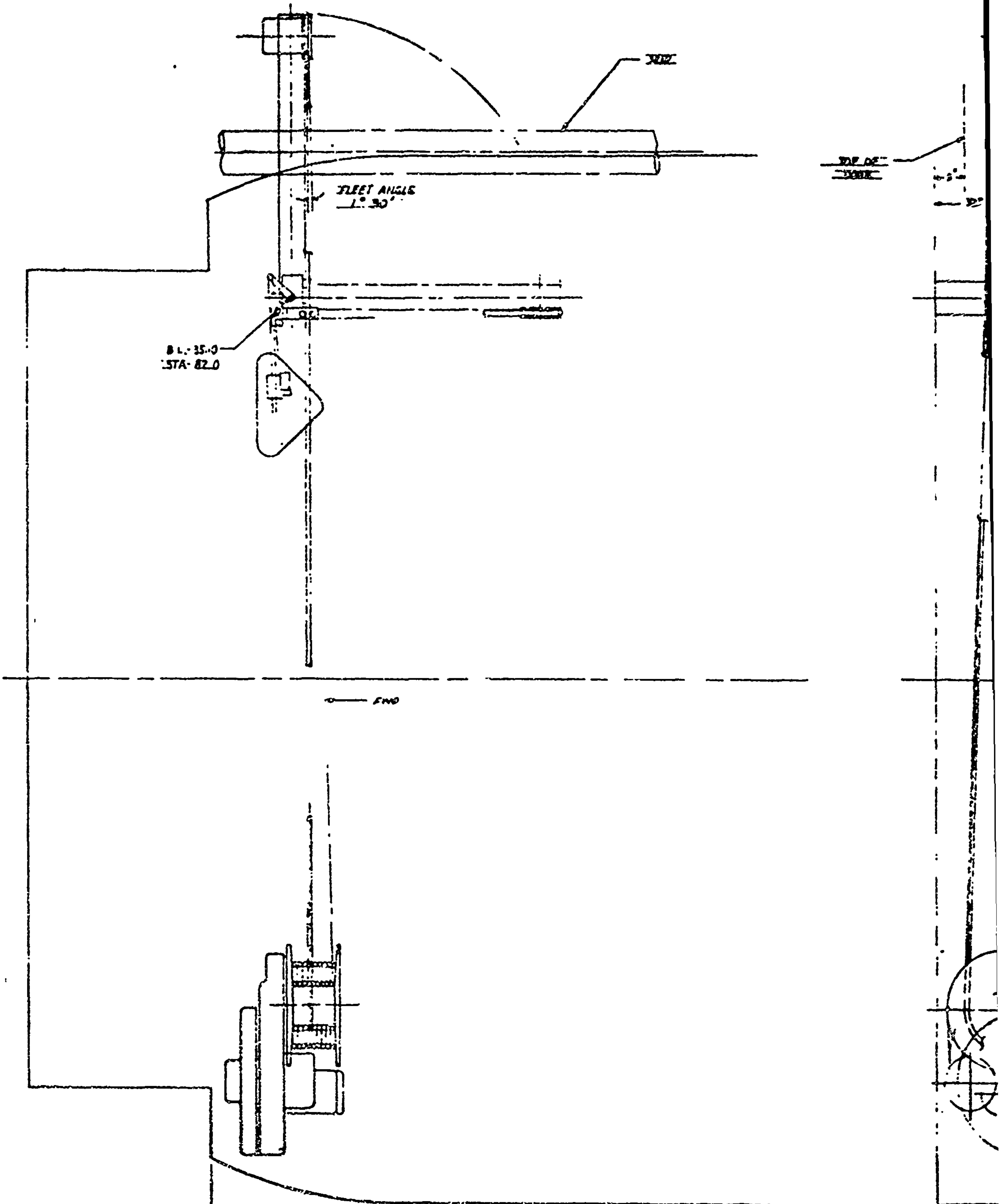

Fig.

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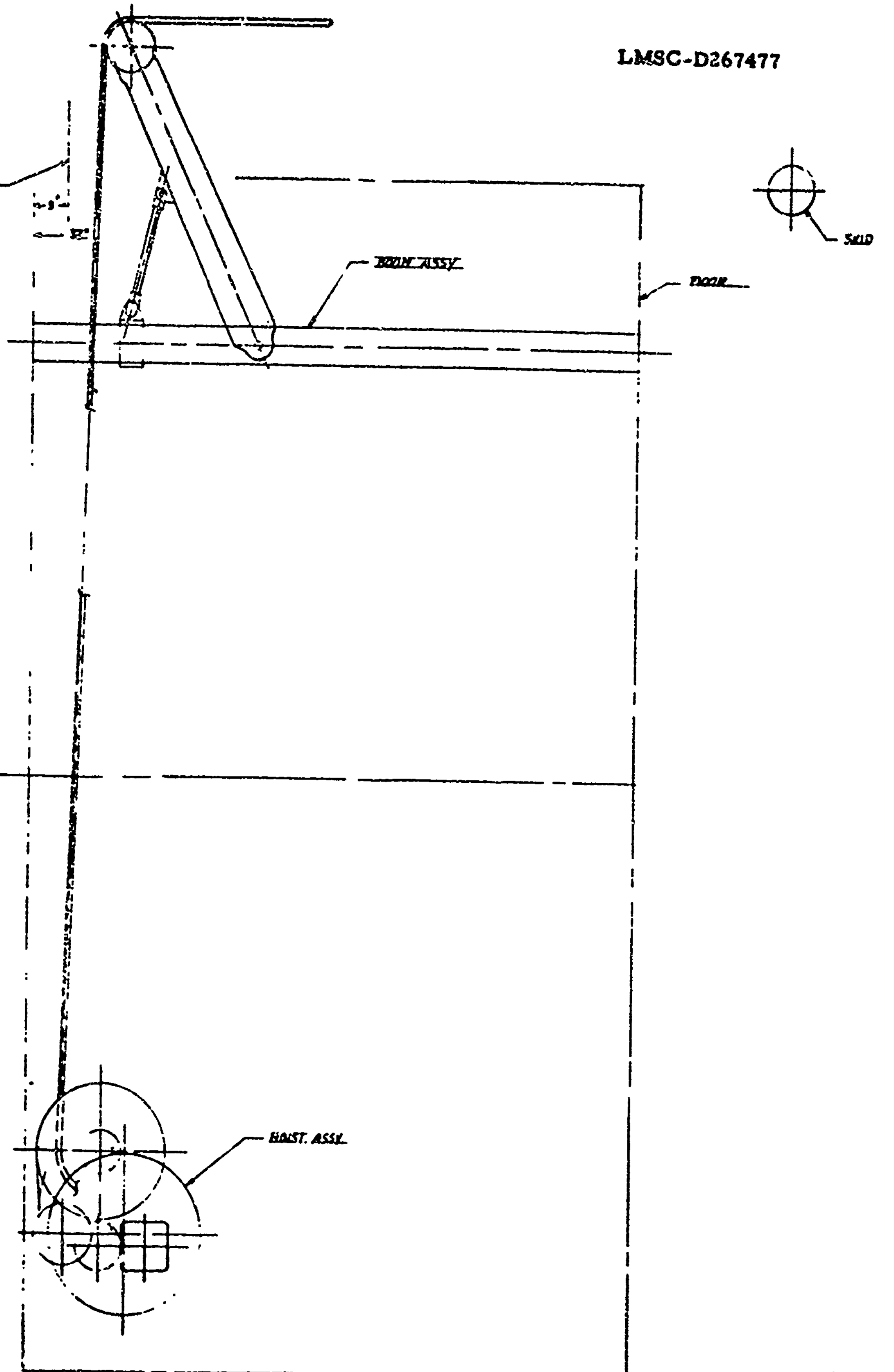

Fig. 7-21 Preliminary Lay ıt - Separate Hoist and Boom Assembly

Section 8
FUTURE PROGRAM PLANS

The following plans are recommended for subsequent program phases, which will provide for the design, fabrication, and testing of a prototype hoist based on the concept selected in the present phase as the most promising. The two subsequent phases described below are in turn intended to be followed by a military potential evaluation. Figure 8-1 depicts the testing aspects of the entire program in terms of verification. The present phase has selected the most promising concept. Phase II in turn verifies that this concept performs under simulated, specific conditions. Phase III verifies that critical components will be satisfactory and that the completed assembly does indeed perform as intended. This is, in turn, followed by a verification that the hoist also performs adequately under actual flight conditions and that this performance is suitable to actual operational conditions.

A schedule for Phases II and III is presented in Fig. 8-2.

PROGRAM PLANNING, DIRECTION, AND COORDINATION

Planning, direction, and coordination will be provided for all aspects of Phase II and Phase III. These tasks include such functions as interface with the Contracting Officer's Representative, schedule and cost control, coordination of task groups, and coordination of Government/Contractor conferences.

PHASE II – FINALIZE HOIST DESIGN CONFIGURATION

In Phase II the necessary verification tests and design analyses will be conducted by the contractor to establish the suitability of the design prior to

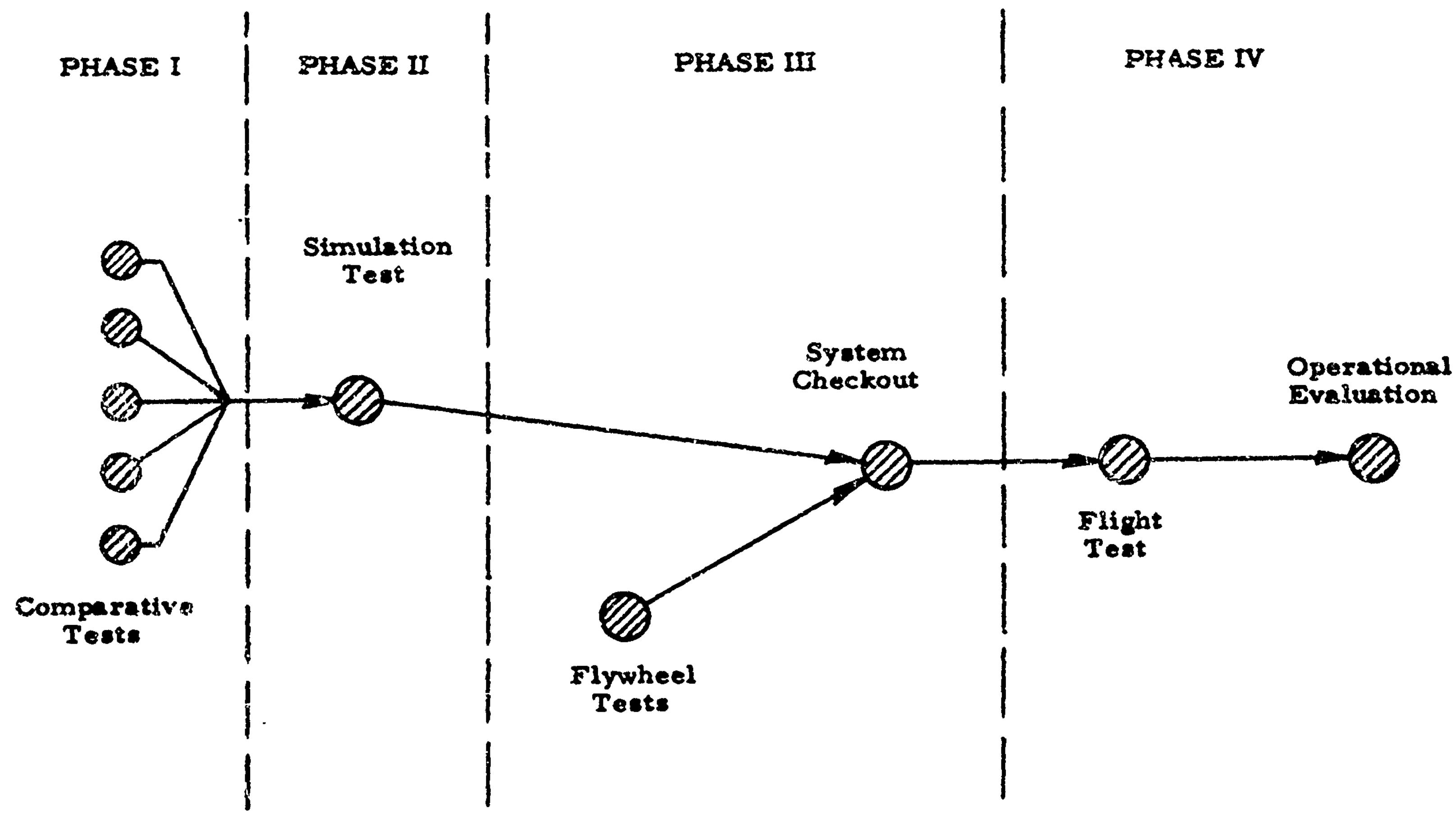

Fig. 8-1 Verification Testing

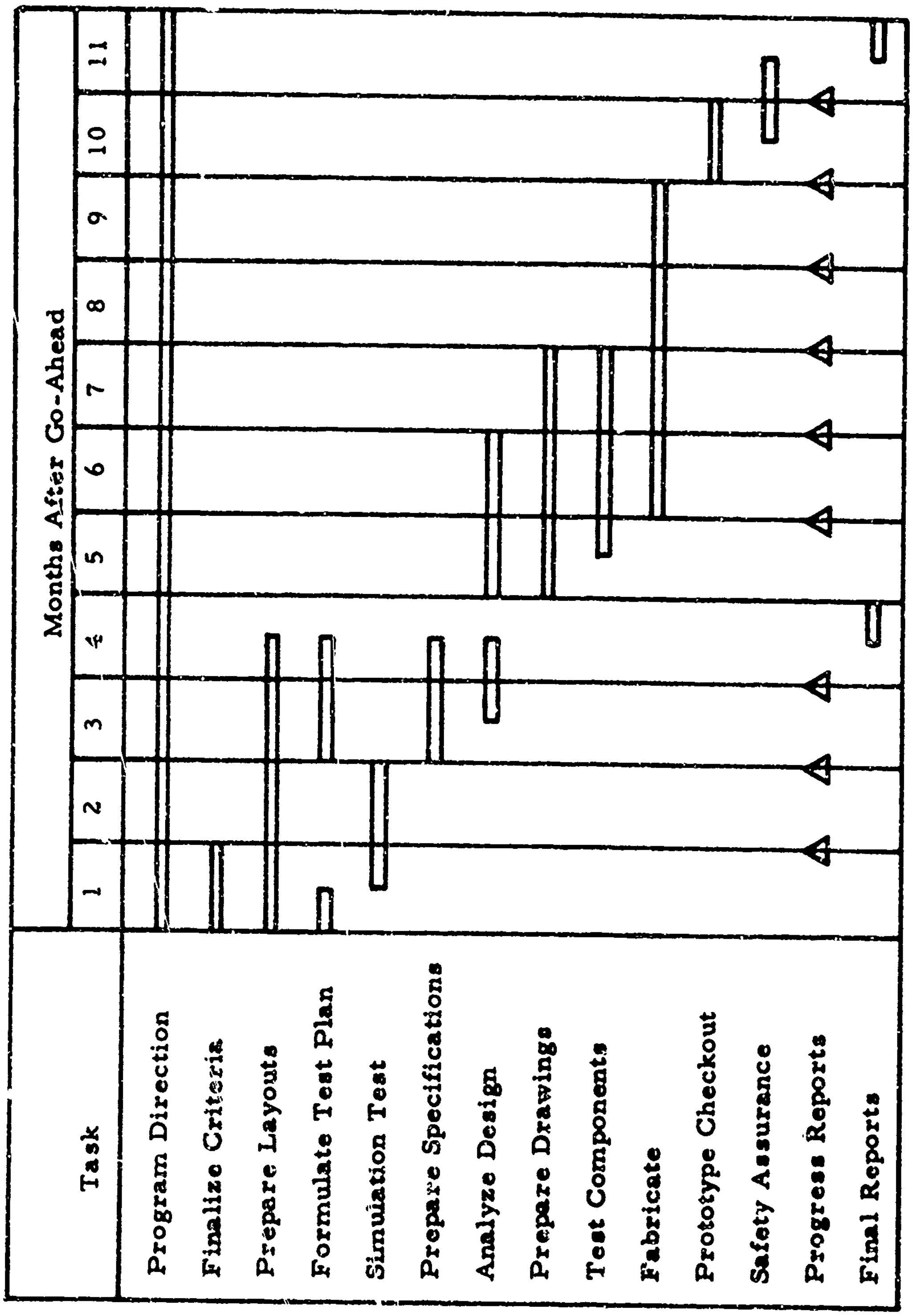

Fig. 8-2 Program Schedule — Phase II and Phase III

the start of the final detailed design in Phase III. Phase II will begin with the contract award and extend for a period of four months.

Finalize Requirements

The configuration previously selected as the most feasible for a high performance hoist will be reviewed (in conjunction with the Contracting Officer's Representative) and the following hoist characteristics will be established as design goals:

- Maximum weight
- Volumetric and dimensional limitations
- Operational duty cycles (with loads and rates)
- Mounting and installing constraints
- Control requirements
- Usable cable length
- Required accessories

Hoist requirements will be finalized, including design factors, and a determination of materials and processes.

Hazard Appraisal

A preliminary appraisal of potential hazards will be made to establish safety assurance test plans. Included will be the areas of:

- Suitability of hoisting rope and handling technique for human use
- Special problems peculiar to flywheels
- Reliability of safety critical components (motors, drives, vacuum pump, etc.)
- Clutch and brake heat dissipation
- Functional suitability of safety devices such as fail safe brake application and control fail-safe modes
- Structural integrity of installation in aircraft
- Effects on stability and control of the aircraft

Simulation Test Plans

Plans for design and fabrication of special equipment will be executed for test program simulating the function of the selected hoist concept.

Simulation Test

A test will be conducted, simulating the geometry, speeds, and loads of the selected preliminary design, to verify the suitability of the chosen hoisting material and method over the prescribed hoist line replacement life. Reliability will be evaluated through a complete spectrum of hoisting conditions including: with full load and unloaded; at full and low speed; under empty, partly reeled and fully reeled conditions; and, with changes of pace at full acceleration rates. Sufficient cycles will be executed to establish abrasion resistance, durability, and lack of degradation with use.

Evaluation will be made of the selected rope pulling technique, means of level winding on storage reel, and the suitability of the rope for high speed reeling. Bending characteristics of the hoisting material will be verified. The use of the descent drive arrangement at high speed will be included.

Prototype Layout

Layouts will be made of the selected hoist design. Vendor designs will be reviewed for component selection. Specifications will be prepared for early procurement of long-lead items. Layouts of control system and installation in aircraft will be included.

Phase III Program Plans

Information necessary to establish program plans for Phase III (which will lead to the military potential evaluation tests, demonstrations, and performance verification of the hoist) will be furnished by the contractor. Firm

program plans will be developed for the final detailed design, fabrication, checkout, and testing (including safety assurance) of a full-size prototype helicopter hoist system suitable for military tests.

PHASE III -- DESIGN, FABRICATE, AND CHECKOUT MILITARY POTENTIAL TEST PROTOTYPE

Phase III will encompass the final design of the prototype hoist, fabrication of one prototype hoist (with spares), and testing of the prototype hoist. This phase will begin with the Contracting Officer's Representative approval and end seven months later.

Test Plans

Plans will be developed for the testing of flywheels and other components. Such tests will include both supplier tests and LMSC tests. Any special equipment which may be required will be designed and fabricated. Additional planning will include functional checkout of the prototype assembly and flight safety assurance.

Component Testing

Tests and inspections will be conducted to verify the safety and performance characteristics of each flywheel. In addition to quality assurance inspection and non-destructive testing to verify conformity to drawings and specifications, an overspeed spin test at 150 percent of design speed will be performed. Vacuum chamber tests will be conducted to establish windage losses.

The reliability and functional suitability of other critical components will be verified by testing by either LMSC or the supplier. These items include the two electric motors, bearings, seals, vacuum pump, brake, and clutch.

Prototype Detail Design

Weight and structural analysis and detail drawings of the hoist prototype layout will be made.

Fabrication

One prototype helicopter hoist, together with spares, will be fabricated and assembled. The quantity and types of spares shall be approved by the Contracting Officer's Representative.

Hoist Assembly Tests

Checkout of completed hoist assembly and its components, under no load, load, and overload conditions, will be followed by functional tests simulating a full spectrum of operations. Both manual and automatic modes will be included, and failure modes simulated to checkout safety devices. Sufficient height will be provided to allow both acceleration to, and deceleration from, full speed. Particular emphasis will be given to brake and clutch performance after repeated cycles.

Flight Safety Assurance Test

Structural integrity of the attachment of the hoist assembly in an Army furnished aircraft will be evaluated. Sufficient tests will be performed to assure that operation of the hoist will not cause the stability and control limitations of the aircraft to be exceeded. Moments generated by gyroscopic precession, lateral unbalance, and dynamic loads will be included.

A functional checkout of the boom extension-retraction system will be made.

Documentation

Brief narrative-type progress/status reports will be presented each month. A final report at the end of Phase III will present test results and conclusions and recommend further activities for military potential evaluation. Preliminary operating and maintenance instructions will be included. The contractor will also prepare and deliver a final briefing to the Government covering the results, conclusions, and recommendations derived from the contract work.

Section 9
CONCLUSIONS AND RECOMMENDATIONS

The overall conclusion of the flywheel-powered High Performance Rescue Hoist Feasibility Study is that flywheel energy storage will provide the power requirements easily permitting a five-fold improvement in lifting speed over the present U. S. Army inventory hoist. Although the specific energy of candidate flywheels studied are not sufficient to make a flywheel-only hoist practical, a flywheel/motor combination will permit the design of a High Performance Helicopter Rescue Hoist without imposing additional power drain upon the helicopter than the existing hoist system.

Specific conclusions and recommendations are as follows:

(1) A flywheel/electric motor combination provides the most efficient, lightweight method of providing large amounts of power necessary in performing high speed rescue hoist operations. A 73.5 w-hr capacity flywheel operated through a 2:1 speed range and continually charged by a 2.0 hp d.c. motor provides enough energy to handle the hoisting requirements encountered in the field at speeds in excess of five times that of the present hoist system.

(2) A typical hoist rescue mission scenario (for design) consists of lifting six 200-lb loads followed by a 600-lb load, all in rapid succession to a height of 210 ft (total weight 1,800 lb) at lifting speeds of 500 ft/min.

(3) The 500 ft per min. lift rate is compatible with the advancement in the "state-of-the-art" sought at this time. Additional increase in speed should be considered in the next generation high speed rescue hoist along with improvements in the man-machine rescue system.

(4) Commutation speed limits of d.c. motors preclude direct motor drive of the flywheel at its optimal speed. A simple spur gear speed increaser for increased shaft speed is the most direct lightweight design for connecting motor to flywheel.

(5) The slip clutch power transmission system offers the simplest, lightweight means of connecting flywheel power to the hoisting mechanism. Hydraulic actuation with electronic controls especially suited to the slip clutch is within the state-of-the-art and readily adaptable to the hoist control requirements.

(6) The optimum flywheel geometry is a modified exponential or conical disc.

(7) The optimum flywheel material is common high-strength steel such as AISI 4340.

(8) The design of steel disc flywheels is simple and straight-forward, and performance is highly predictable. Manufacturing, inspection, and test techniques are well established for similar hardware such as turbine rotors.

(9) Gyrodynamic effects of the flywheel on the control of the helicopter is negligible.

(10) Design of flywheel with adequate safety margins will preclude flywheel destruction from rotative forces.

(11) Plastic braided rope has many properties making it the prime candidate for use on the high speed rescue hoist.

(12) Additional studies and tests are recommended to verify and augment the findings on rope characteristics and rope handling techniques of this program.

Program plans for the development and test of a High Performance Helicopter Rescue Hoist are given in Section 8, Future Program Plans.

Section 10
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APPENDICES

Appendix A1
FLYWHEEL WINDAGE LOSSES

For loss, in terms of steady-stage windage:

$$hp = 0.006\ (1 + \frac{2.3}{Ro})\ (\frac{P}{T})^{0.8}\ (\frac{N}{10^4})^{2.8}\ (Ro)^{4.6}\ (\mu)^{0.2}$$

where:

- t = Flywheel tip thickness = 0.343 in.
- R = Flywheel outer radius = 6.0 in.
- P = Flywheel housing press. = 0.097 psia
- T = Flywheel housing air temperature = 520°R
- N = Flywheel rotational speed = (N) rpm
- μ = Air viscosity = 0.0431 lb/hr-ft

$$hp = 0.006\ (1 + \frac{2.3 \times 0.343}{6.0})\ (\frac{0.097}{520})^{0.8}\ (\frac{N}{10^4})^{2.8}\ (6.0)^{4.6}\ (0.0431)^{0.2}$$

$$hp = 19.25\ (\frac{N}{10^4})^{2.8} \times 10^{-3}$$

Losses for various speeds are tabulated below:

N RPM	$(\frac{N}{10}\ 4)^{2.8}$	HP Loss
32,000	26	0.500
28,000	17.8	0.342
24,000	11.6	0.223
20,000	6.95	0.134
16,000	3.73	0.072

Evaluating the effects of varying pressures at 28,000 rpm:

$$hp = C\,P^{0.8} = 0.342$$

$$C = \frac{0.542}{(0.097)^{0.8}} = \frac{0.342}{0.154} = 2.22$$

Losses for pressure levels 0 - 30 mm Hg are tabulated below:

Pressure Hg	Pressure P PSIA	$P^{0.8}$	HP
0	0	0	0
5	0.097	0.154	0.342
10	0.193	0.268	0.594
15	0.290	0.371	0.823
20	0.386	0.467	1.035
30	0.580	0.647	1.435

Appendix A2
FLYWHEEL BEARING CALCULATIONS

Principal bearing loads consist of centrifugal force and precession loads.

For centrifugal force loads (CF):

Data

r = displacement of CG, 16.7×10^{-6} ft (0.0002 in.)
Flywheel weight, $W = 13.50$ lb
Speed, $N = 28,000$ rpm, $N^2 = 7.84 \times 10^8$

Calculations

$$
\begin{aligned}
CF &= 3.41 \times 10^{-4}\, W r N^2 \\
&= (3.41 \times 10^{-4})(13.5)(16.7 \times 10^{-6})(N^2) \\
&= 7.68 \times 10^{-8} N^2 \\
&= (7.68 \times 10^{-8})(7.84 \times 10^8) \\
&= 60.2 \text{ lb}
\end{aligned}
$$

For precessional torque load, Tp:

$$Tp = I_o \omega \Omega$$

Data

I_o = 0.045 ft-lb sec^2
ω = Flywheel operating speed, rad/sec = 2,930 rad/sec
Ω = Precessional rotative speed, rad/sec = 0.0 rad/sec

Calculations

$$Tp = 0.045 \times 2,930 \times 1.0 = 132 \text{ ft-lb}$$

Tp for various precession rates are tabulated below:

N RPM	W Rad/Sec	1.5	1.0	0.8	0.6	0.4	0.2
		Rad/Sec — Precession Torque, ft-lb					
16,000	1,675	113	75	60	45	30	15
20,000	2,090	140	94	75	56	38	19
24,000	2,510	170	113	90	68	45	23
28,000	2,930	198	132	105	80	53	27
32,000	3,350	226	150	120	90	60	30

Flywheel Assembly Bearing Life Analysis

The L_{10} rating for the bearing is calculated as follows:

$$*L_{10} = \frac{50,000}{N} \left(\frac{C_B}{R_E}\right)^3$$

where

N = rpm

C_B = basic radial load rating at 33-1/3 rpm

R_E = equivalent radial load

L_{10} = life (hr)

The equivalent L_{10} life of a bearing subject to varying speeds for varying times can be determined by the formula:

$$L = \frac{1}{\frac{P_1}{L_1} + \frac{P_2}{L_2} \cdots + \frac{P_N}{L_N}}$$

*L_{10} life is defined as the number of hours (at some given constant speed and load) that 90 percent of a group of bearings will complete or exceed before the first evidence of fatigue develops. Average life is approximately five times the L_{10} life as presently determined (10).

where

L = equivalent hours of L_{10} life

P_1 = portions of time expressed as a decimal fraction of time that load and speed are in effect

L_1 = calculated life of each bearing at each load and speed

Combining the steady centrifugally induced loads with the precession forces and calculating the L_{10} life for each condition results in the L_{10} lives are as shown below:

Cond.	Percent of Time	Normal Loading Radial	Gyro Induced Loads	Total Loads Radial	Theoretical Life - L_{10} (Hr)
1	0.0001	36.8	594	630.8	24
2	0.0005	36.8	396	432.9	74
3	0.005	36.8	316	352.9	132
4	0.025	36.8	238	274.9	292
5	0.033	36.8	159	195.9	810
6	0.055	36.8	79.5	116.4	3,690
7	0.88	36.8	-	36.85	120,000

Combined L_{10} bearing life for duty cycle = 7,000 hr

Bearing Drag Loss Calculation

Bearing Data

Brg 9104 (20 x 42 x 14)

Contact angle = 12 deg

Rotational speed = 28,000 rpm

$$\text{Friction Torque} = 0.083\, f_1 P_B d_m + 1.183 \times 10^{-6} f_o\, (vN)^{2/3}\, d_m^{\ 3} \quad \text{(Ref. 10, pg 46)}$$

$$\text{Friction Torque} = (0.083)(0.000018)(36.8)(1.22) + 1.183 \times 10^{-6}(1.5)(3,700)(1.8)$$

$$= \underline{0.012} \text{ in. lb}$$

$$\text{hp loss} = \frac{(0.012)(28,000)}{63,000} = 0.00534 \text{ hp/brg}$$

Appendix A3
FLYWHEEL CHAMBER VACUUM PUMP REQUIREMENTS

Size Calculations:

Data

Chamber vol., $V = 0.014\ ft^3$
Ambient pressure, $P_1 = 760$ mm Hg
Chamber pressure, $P_2 = 5$ mm Hg
Pump down time, $\Delta T = 0.5$ min.
Outgassing load, Q_o = Assume 0.1 Torr cfm (see Fig. A3-1)
Leakage load, Q_L = 15.2 Torr-cfm
Conductance = 1,000 (see Fig. A3-2)

Calculations

Pump capacity:

$$\text{cfm (load)} = \frac{2.3}{\Delta T} V \log\left(\frac{P_1}{P_2}\right) + \frac{Q_o + Q_L}{P_2}$$

$$= \frac{2.3}{0.5}(0.014) \log\left(\frac{760}{5}\right) + \frac{0.1 + 15.2}{5}$$

$$= 0.14 + 3.06 = 3.20 \text{ cfm}$$

Since conductance is negligible, pump capacity = pump load = 3.20 cfm

Pump power requirements (from Fig. A3-3):

Power consumption = 0.17 w-hr/ft^3

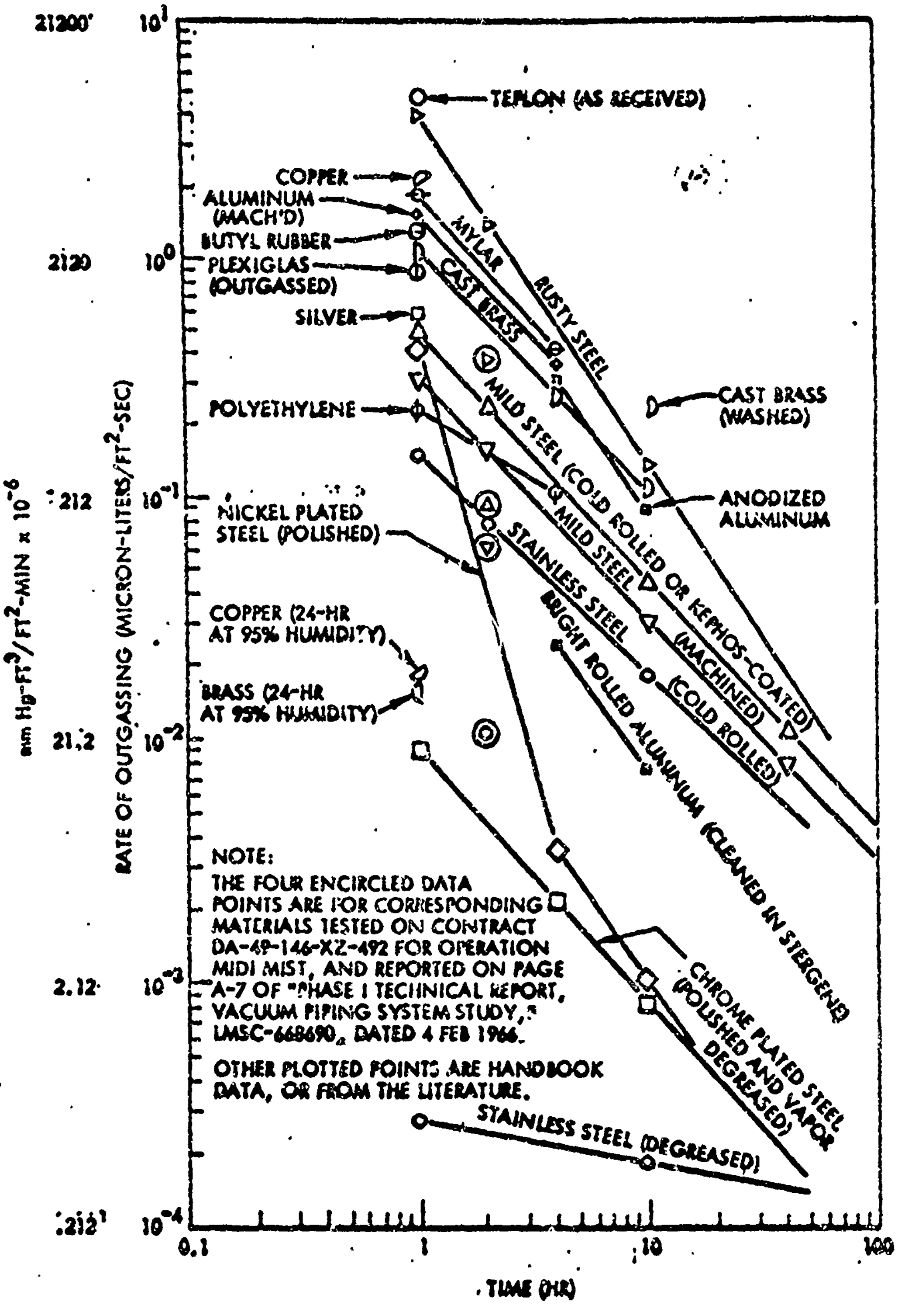
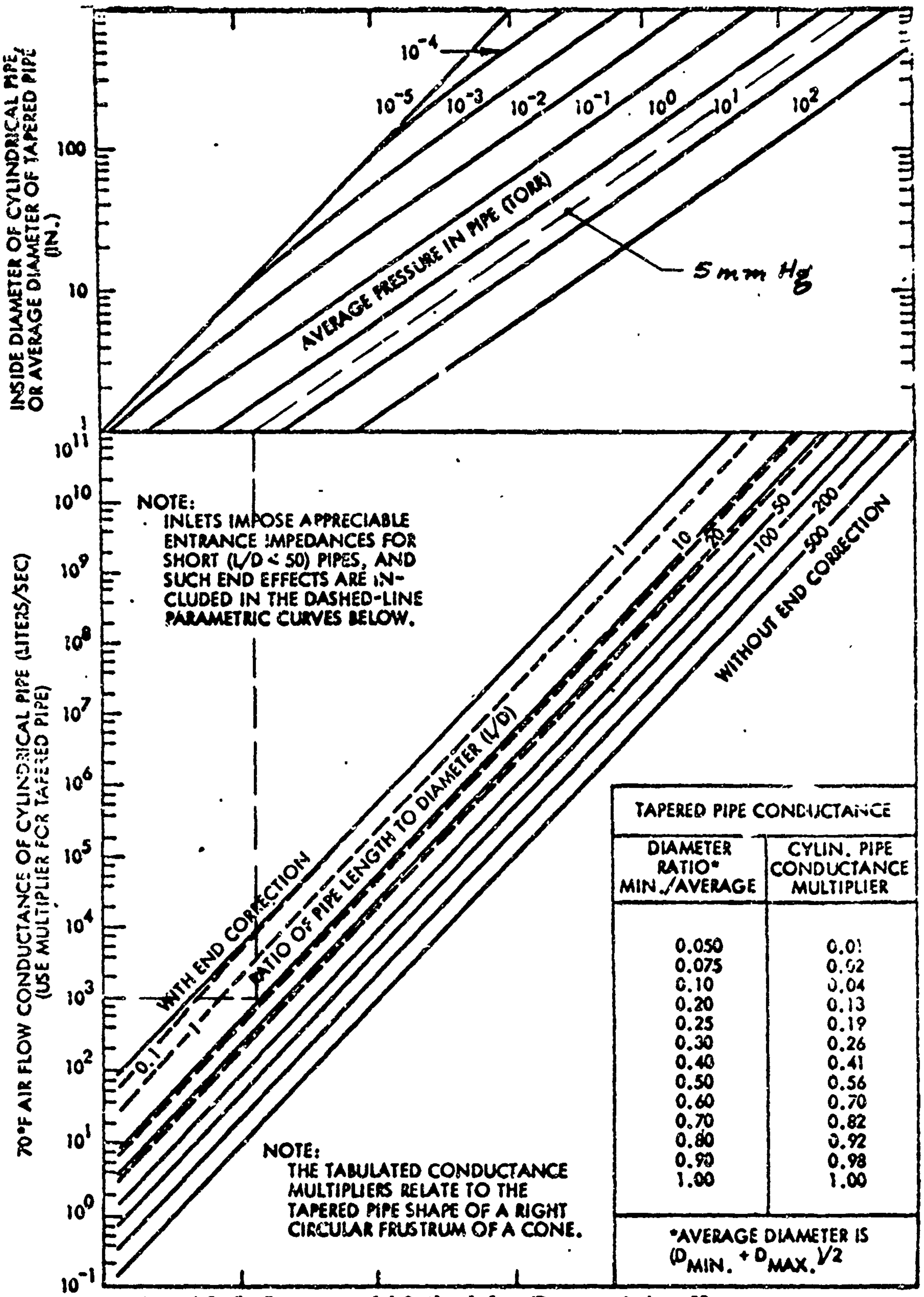


Fig. A3-1 Outgassing Rate Comparison

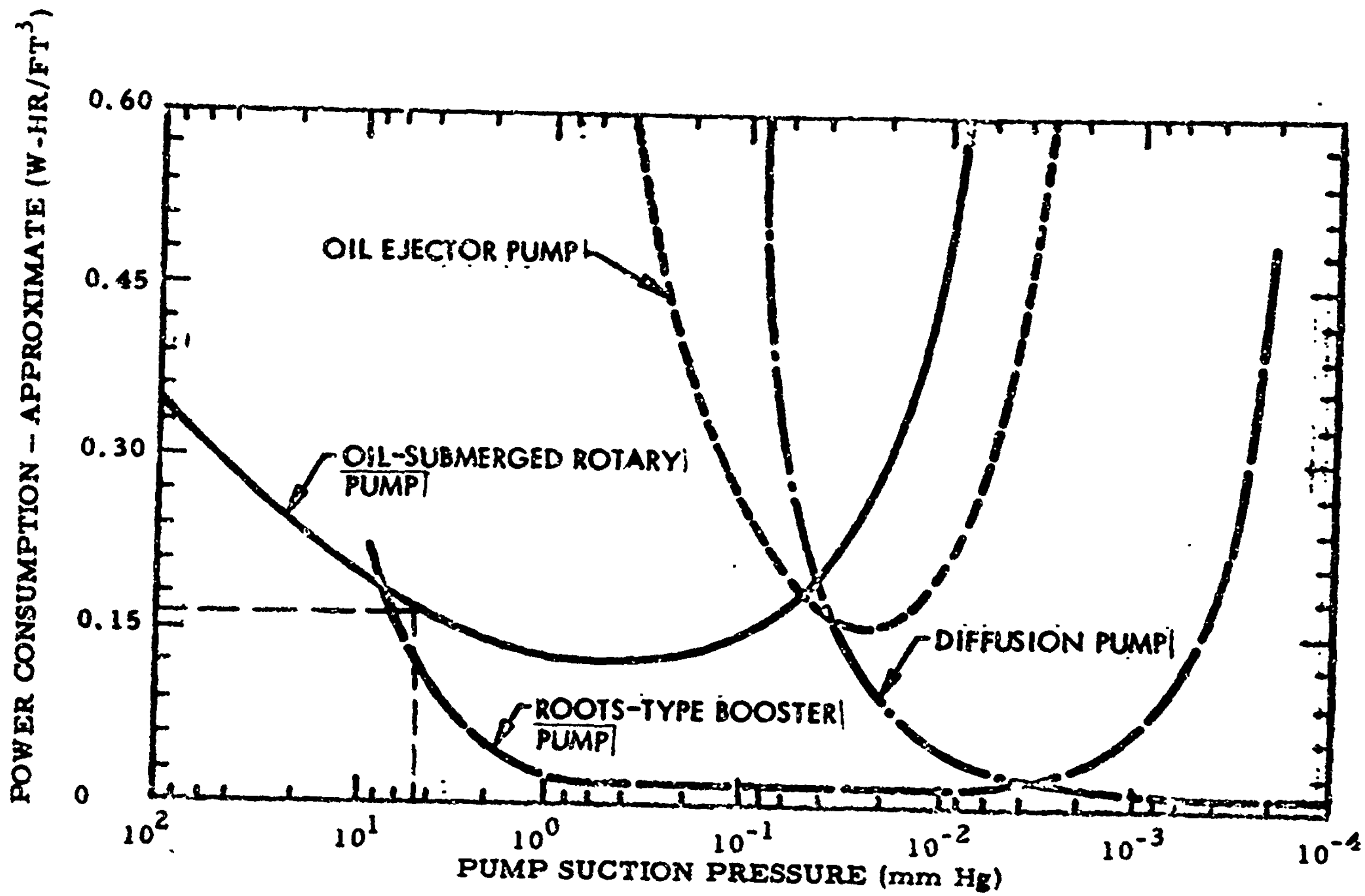


TAPERED PIPE CONDUCTANCE

DIAMETER RATIO* MIN./AVERAGE	CYLIN. PIPE CONDUCTANCE MULTIPLIER
0.050	0.01
0.075	0.02
0.10	0.04
0.20	0.13
0.25	0.19
0.30	0.26
0.40	0.41
0.50	0.56
0.60	0.70
0.70	0.82
0.80	0.92
0.90	0.98
1.00	1.00

*AVERAGE DIAMETER IS $(D_{MIN.} + D_{MAX.})/2$

Fig. A3-2 Improved Method for Determining Vacuum Pipe Flow Conductance



POWER CONSUMPTION IS BASED UPON PUMPING CAPACITY OF THE PUMP, IN FT^3, AT THE SUCTION PRESSURE CONCERNED.

Fig. A3-3 Power Consumption of Various Pump Types Relative to the Working Pressure

Since:

$$3.20 \text{ cpm} \times 60 = 192 \text{ cfh}$$

$$192 \frac{ft^3}{hr} \times \frac{0.17 \text{ w-hr}}{ft^3} = 32.6 \text{ watts}$$

For hp:

$$32.6 \text{ w} \times 1.341 \frac{hp}{w} \times 10^{-3} = 0.044 \text{ hp}$$

Pump efficiency is approximately 20 percent due to low working pressure.

$$\text{Power load is } \frac{0.044}{0.20} = 0.22 \text{ hp}$$

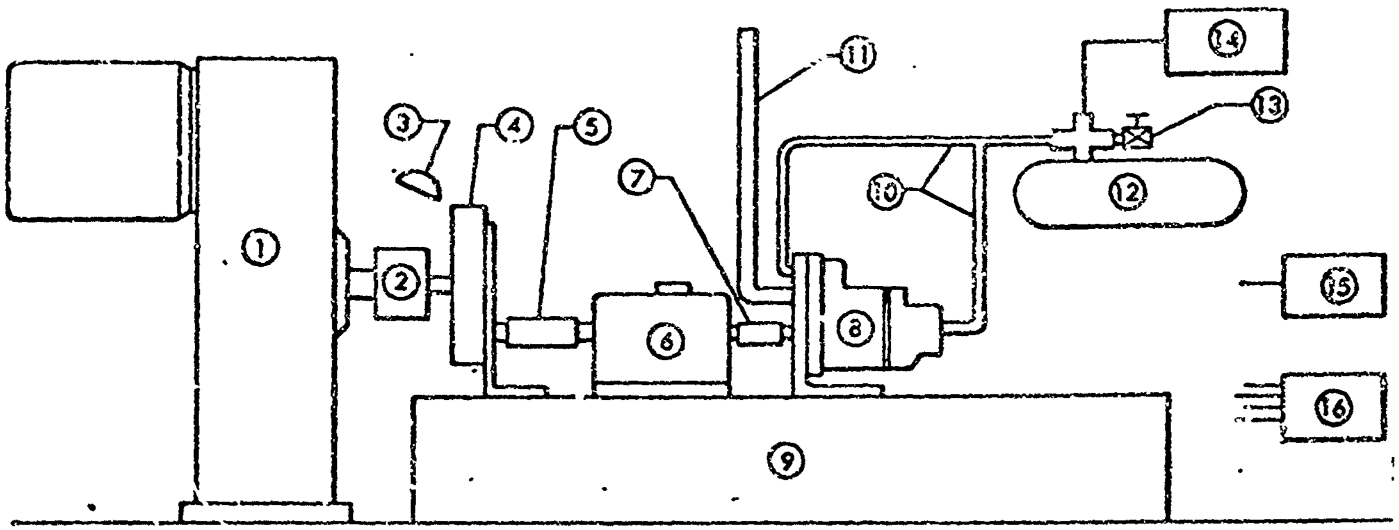
Power requirements for chamber pressure ranging to 30 mm Hg are tabulated below:

P_2 mm Hg	$\frac{P_1}{P_2}$	$Lg \frac{P_1}{P_2}$	$0.063\ Lg \frac{P_1}{P_2}$	$\frac{15.3}{P_2}$	CFM	HP
5	152.0	2.18	0.138	3.06	3.20	0.22
10	76.0	1.88	0.119	1.53	1.65	0.112
15	50.6	1.71	0.108	1.02	1.13	0.077
20	38.0	1.58	0.100	0.76	0.86	0.059
30	25.3	1.40	0.089	0.51	0.60	0.041

VACUUM PUMP VERIFICATION TESTS

Test Objective

The test objective was to determine the suitability of a gerotor-type oil pump for use as a vacuum pump on high-speed flywheel applications. The test setup for this test is shown by Fig. A3-4 (pump test setup).



1 Varidrive Unit
2 Waldron Coupling (grease lubricated)
3 Strobe-Tachometer
4 Speed Increaser Gearbox (Recirculating oil lube)
5 Splined Coupling (Grease lubricated)
6 Torque Sensor, Lebow Model 1102-200 (0 – 200 In.-Oz Cap). Dead Weight Calibrated and Air-Oil Mist Lubricated
7 Coupling (Grease lubed)
8 Test Vacuum Pump
9 Base
10 Plumbing – Parallel System From Vacuum Chamber to Inlets of Dual Elemet Pump
11 Standpipe – Outlet Port of Pump; Minimum Oil Height = 12.0 In.
12 Vacuum Chamber
13 Vacuum Shutoff Valve
14 Vacuum Gage
15 Daytronic Model 770 Strain Indicator
16 Temperature Recorder (Monitors pump & bearing temps)

Fig. A3-4 Pump Test Setup

Test Procedure

The pump was tested in three different configurations to determine its best arrangement for use as a vacuum pump. The configurations are as follows:

- Test 1 consisted of using one element as a vacuum pump and the second element as a scavenge pump.
- Test 2 consisted of using both elements as a vacuum pump and lubricating the pump with an oil reservoir which was attached to the discharge port.
- Test 3 consisted of using one element as a vacuum pump and removing the other element. Lubrication was accomplished in the same way as in Test 2.

Test Results

Test results are shown in Tables A3-1, A3-2, and A3-3, and the pump flow curve is shown in Fig. A3-5.

Conclusions

Test results show the pump to be suited to the flywheel system for the following reasons:

- Pump downtime to a useful vacuum level never exceeded 25 sec. A typical pump downtime curve is plotted in Fig. A3-5.
- The pump is capable of pumping down and holding air pressure levels below 5 mm Hg.
- Configuration 1 provides the highest vacuum producing capability.
- Temperature stabilized at acceptable levels for all configurations.

Table A3-1

VACUUM PUMP TEST 1

Date 1/20/72

Recorder M. Helvey

Witness R. Ruth

Type of Pump Gerotor Scavenge Pump

Pump Identification GC 436 M

Pump Configuration 1 Element Oil Pump

1 Element Vacuum Pump

Vacuum Chamber Volume 108 in.^{3}

Ambient Pressure 29.92 psi

Ambient Temperature 68°F

Item	Pump Speed (rpm)			
	5,200	6,000	7,000	8,000
Pumpdown Time (sec)	35	40	40	30
Minimum Pressure Attained (mm Hg)	1.9	2.8	2.9	5.1
Pump Housing Temperature (°F)	100	175	175	215
Torque (in.-oz)	Off Scale; Test Setup Records only up to 100.00 in.-oz			

Table A3-2

VACUUM PUMP TEST 2

Date 1/20/72 Recorder M. Helvey

Witness R. Ruth

Type of Pump Gerotor Scavenge Pump

Pump Identification P/N GC 436 M

Pump Configuration Both Elements Vacuum Pump

Vacuum Chamber Volume 108 $in.^3$

Ambient Pressure 29.92 psi

Ambient Temperature 68°F

Item	Pump Speed (rpm)			
	5,200	6,000	7,000	8,000
Pumpdown Time (sec)	37	32	20	17.5
Minimum Pressure Attained (mm Hg)	17.9	21.0	95.0	190.0
Pump Housing Temperature (°F)	148	180	210	210
Torque (in.-oz)	93.02	79.02	66.70	57.40
Horsepower	(0.48)	(0.47)	0.464	0.455

Table A3-3

VACUUM PUMP TEST 3

Date 1/20/72

Recorder M. Helvey

Witness R. Ruth

Type of Pump Gerotor Scavenge Pump

Pump Identification P/N GC 436 M

Pump Configuration 1 Element Vacuum Pump

1 Element Removed

Vacuum Chamber Volume 108 in.^{3}

Ambient Pressure 29.92 psi

Ambient Temperature 68°F

Item	Pump Speed (rpm)			
	5,200	6,000	7,000	8,000
Pumpdown Time (sec)	31	25	20	30
Minimum Pressure Attained (mm Hg)	9.5	34	39	30
Pump Housing Temperature (°F)	140	156	160	210
Torque (in.-oz)	46.0	41.7	40.5	31.7
Horsepower	(0.237)	(0.248)	0.281	(0.252)

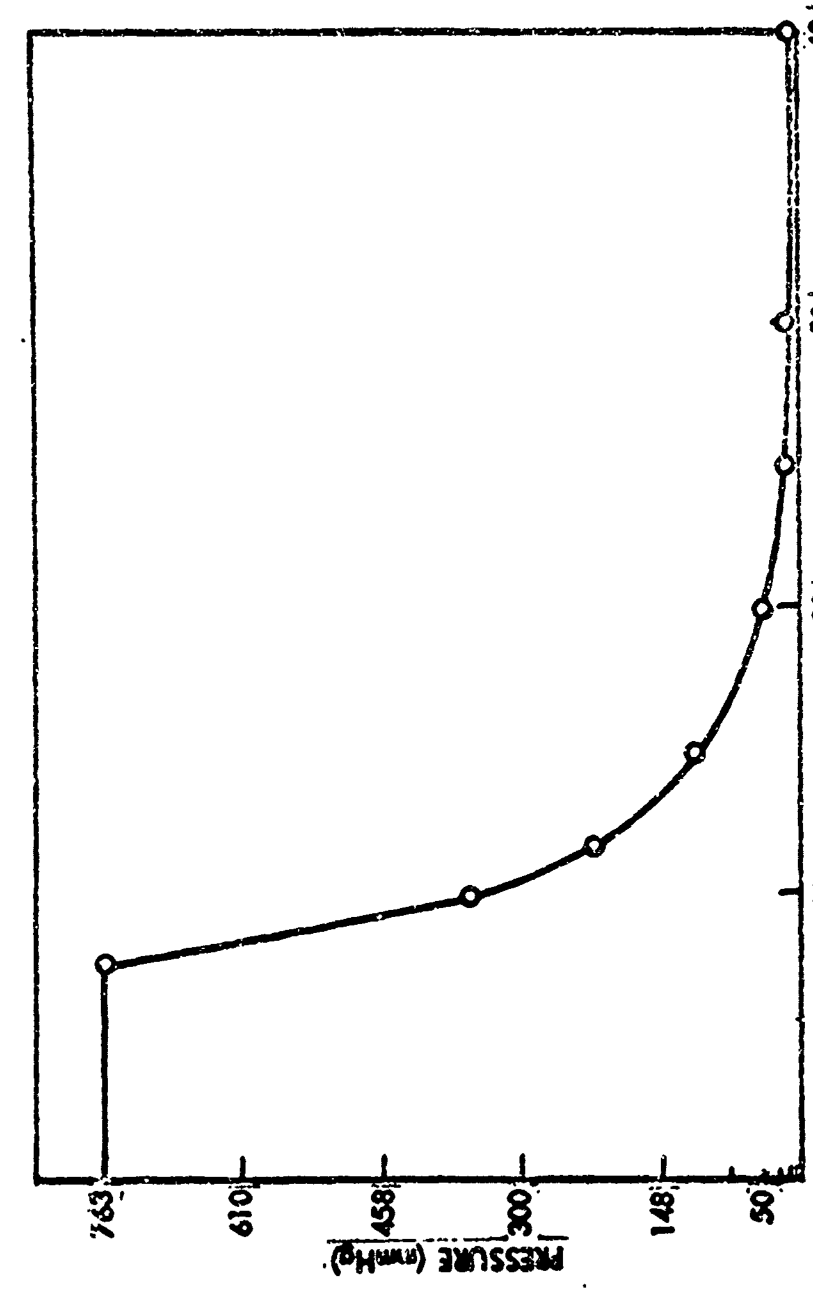


Fig. A3-5 Pump Down-Time – Gerotor Pump

Appendix A4
GEAR ANALYSIS – FIRST REDUCTION STAGE

Gear Data

Ratio = 2:1

P_d	= 19	F	= 0.38
N_p	= 19	N_g	= 38
d_p	= 1.00	D_g	= 2.0
n_p	= 28,000 rpm	n_g	= 14,000 rpm
J	= 0.32 for $\phi = 20^\circ$		

The allowable power formula for a gearset based on tooth strength is:

$$P_{at} = \frac{n_p\, d\, F\, J\, S_{at}}{126,000\, P_d} \cdot \frac{K_v\, K_l}{K_o\, K_m\, K_s\, K_t\, K_r} \qquad \text{(Ref. 13)}$$

where

P_{at} = power, hp
n_p = pinion speed, rpm
d = pinion oper. pitch diam., in.
F = face width
J = tooth geometry factor = 0.32
S_{at} = allowable stress = 33,000 psi
P_d = diametral pitch
K_v = velocity factor = 1.0
K_l = life factor = 1.0
K_o = overload factor = 1.0
K_m = load dist. factor = 1.25
K_s = tooth size factor = 1.0
K_t = temp. factor = 1.0
K_r = safety factor = 1.5

Then

$$P_{at} = \frac{28,000 \times 1.0 \times 0.38 \times J \times 33,000}{126,000 \times 19} \cdot \frac{1.0 \times 1.0}{1.0 \times 1.25 \times 1.0 \times 1.0 \times 1.5} = 25 \text{ hp}$$

The allowable power formula for a gearset based on surface durability is:

$$P_{ac} = \frac{n_p F}{126,000} \cdot \frac{I\, C_v}{C_s\, C_m\, C_f\, C_o} \left(\frac{S_{ac}\, d}{C_p} \cdot \frac{C_l\, C_h}{C_t\, C_r} \right)^2 \quad \text{(Ref. 14)}$$

where

P_{ac}	= power, hp		C_o	= overload factor = 1.0
n_p	= pinion speed, rpm		C_l	= life factor = 1.0
F	= face width, in.		C_h	= hardness factor = 1.0
I	= tooth geometry factor = 0.104		C_p	= material factor = 2,300
C_v	= velocity factor = 1.0		C_t	= temp. factor = 1.0
C_s	= tooth size factor = 1.0		C_r	= reliability factor = 1.25
C_m	= mounting factor = 1.3		S_{ac}	= allowable contact stress = 110,000 psi
C_f	= surface condition factor = 1.0		d	= pinion pitch 'am. = 1.00

Then

$$P_{ac} = \frac{28,000(0.38)}{126,000} \times \frac{0.104 \times 1}{1 \times 1.3 \times 1 \times 1} \times \left[\frac{110,000 \times 1}{2,300} \cdot \frac{1.0 \times 1.0}{1.0 \times 1.25} \right]^2$$

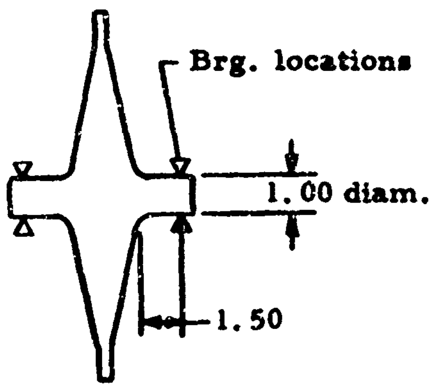
$= 15.2$ hp <u>Marginal Design</u>

Redesign using hardened, ground gears with S_{ac} equal to 180,000 psi (Rc 55 min) (ref. 14), P_{ac} then becomes:

$$P_{ac} = 15.2 \left(\frac{180,000}{110,000} \right)^2 = 40.4 \text{ hp}$$

Smaller gearsets are possible to carry the required nominal maximum power requirement of 20 hp. However, sizes of other components such as motors, clutches, flywheel, brake. etc. determine minimum center distances, thereby governing pitch diameters. Select face widths to suit the power requirements without going to impractically thin gears.

Appendix A5
FLYWHEEL CRITICAL SPEED ANALYSIS



Flywheel Data

Wt = 13.50

Material Steel:

$E = 30 \times 10^6$

$I = 0.0491$

$M = \dfrac{13.50}{386} = 0.35$

ω = critical speed, rad/sec

$$k = \frac{3EI}{\ell^3} = \frac{3 \times 30 \times 10^6 \times 0.0491}{(1.5)^3}$$

$$= 1.3 \times 10^6 \text{ lb/in. (one side)}$$

Assuming support bearing and housing stiffness to be 1×10^6 lb/in., stiffness at each bearing is:

$$k = \frac{1}{\frac{1}{k_1} + \frac{1}{k_2}} = \frac{1}{\frac{1}{1.305 \times 10^6} + \frac{1}{1 \times 10^6}} = 0.51 \times 10^6 \text{ lb/in.}$$

$$K_T = 2 \times k = 1.02 \text{ (for both bearings)}$$

The critical speed formula is:

$$\omega = \sqrt{\frac{K_T}{M}}$$

where

ω = critical speed, rad/sec

k_T = spring rate of supporting structure, $\frac{\text{lb}}{\text{in.}}$

M = mass of flywheel $\frac{\text{lb sec}^2}{\text{ft}}$

Then

$$= \frac{1.02 \times 10^6}{0.035} = 5,380 \text{ rad/sec}$$

or

51,400 rev/min.

Appendix B
EVALUATION TESTS
CANDIDATE HOIST CABLE PROPERTIES

TEST PROGRAM DESCRIPTION

An extensive study and test program was conducted to evaluate candidate lifting materials other than the conventional wire rope. Materials and configurations considered provided certain advantages to the High Performance Hoist design concept which made an investigation into their properties necessary.

Properties investigated were:

- o Elongation
- o Permanent deformation
- o Breaking strength
- o Damping characteristics
- o Self-level winding capabilities
- o Abrasion resistance

Two plastic rope types and one swaged wire cable were compared with the conventional 19 x 7 CRES non-rotating wire cable (MIL-W-83140). One of the plastic rope types called Samson 2 in 1 stable braid has a polypropylene inner braid and a polyester outer braid; the other plastic rope type considered was webbing of a polyester material, the wire cable considered was an American Chain & Cable Co. manufactured 19 x 7 swaged CRES cable of 0.208 diam.

Elongation values were obtained by measuring the change in gage length of each specimen as a tensile load was applied. The property of a permanent deformation was noted by the amount of set the rope had after load was released.

In the same test manner the ropes were loaded until failure occurred; this value is the breaking strength of the material. (See Fig. B-1.) Two candidates, the plastic braid and the wire rope, failed at the manufacturer's stated value; however, the polyester webbing sustained an overload of 25 percent without failing. This overload was at the test machine's load limit so it was not possible to break the specimen with the available equipment.

Damping characteristics were investigated in a quantitative manner, here again, compared to the conventional wire rope. For this series of tests approximately 75 ft of rope was attached to a hoist and used to lift a weight of 217 lb. This weight was then torsionally rotated a preset number of turns and released. Time and oscillations were observed as the load came to zero test position. This gave information of the rope's torsional stiffness. To evaluate the longitudinal (axial) stiffness a weight was manually lifted and released; the time to come to rest was a measure of stiffness in the longitudinal direction. These stiffness tests were then reported with human subjects.

High speed tests were run on the Breeze hoist to determine the characteristics of the capstan type drive at high line speeds. A "Varidrive" unit was substituted for the existing hoist drive motor, which provided a speed capability equivalent to over 500 ft/min. line speed. (See Fig. B-2.)

Self-level wind ability was determined for the plastic braid rope and the wire cable. The webbing material was not tested since the design concept for this material did not require it to have level-wind capability. The method used to evaluate the rope's self-level wind capability was to pull the rope with a powered sheave over a fairlead pulley against a constant load. It was possible to vary the sheave speed up to a rope speed of 1,000 fpm; the flange spacing of the sheave was also adjustable to 6 in. By providing for adjustable center-distance spacing of the fairlead pulley, fleet angles could be changed to accommodate the winding characteristics of each candidate rope. Constant load source for the test was provided by pulling against the

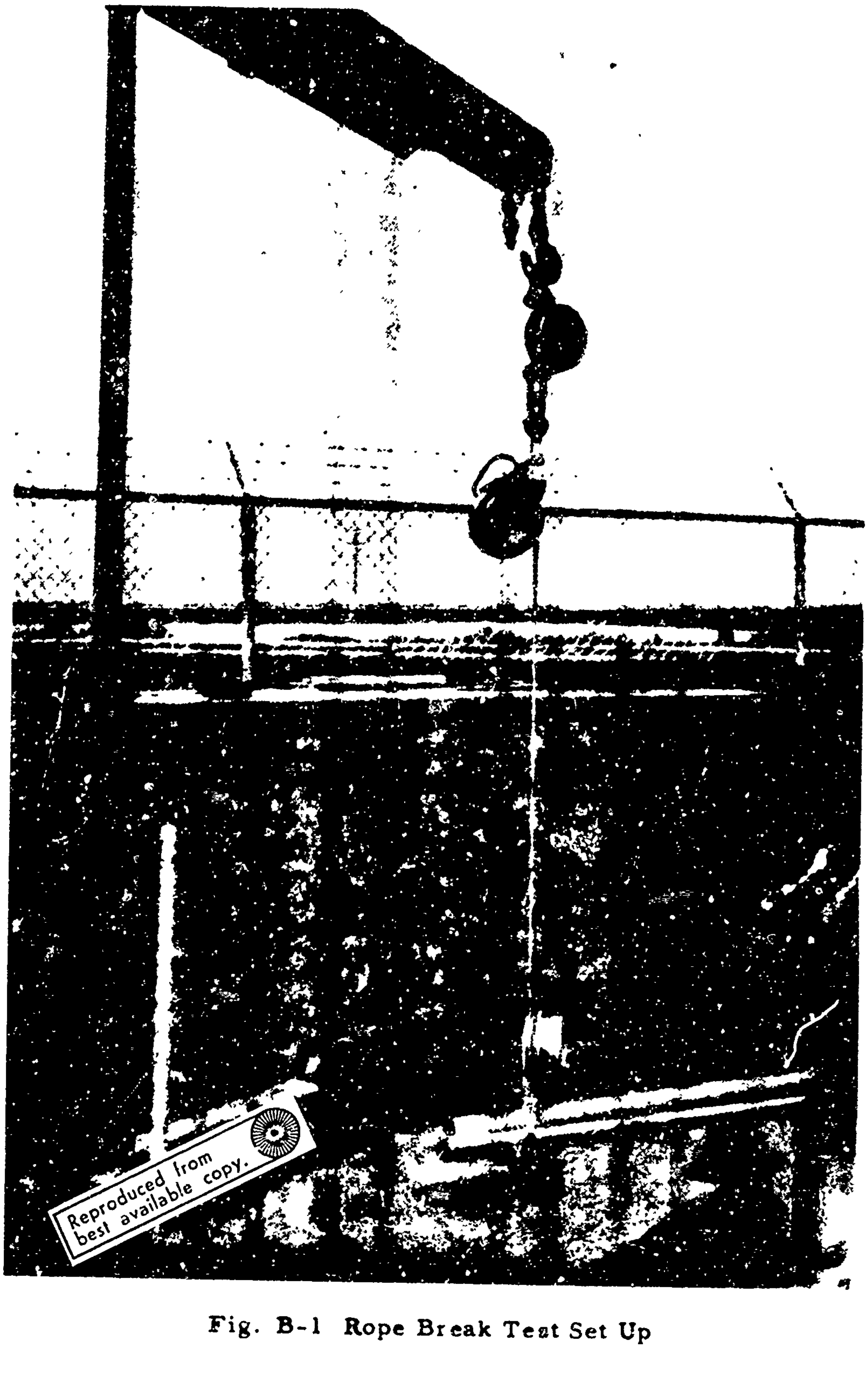


Fig. B-1 Rope Break Test Set Up

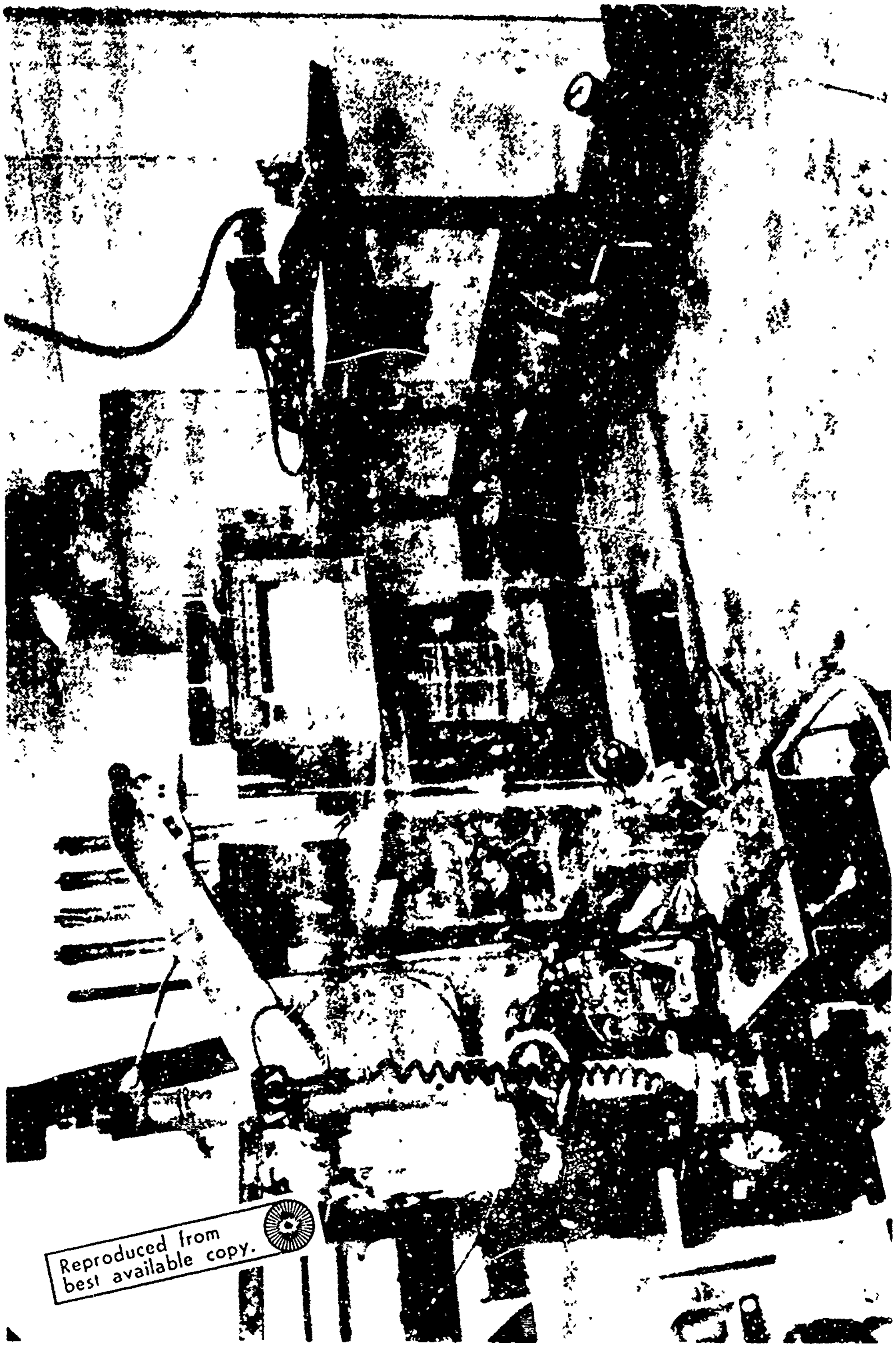


Fig. B-2 High Speed Capstan Test

rolling resistance of a 2-1/2-ton truck. (See Figs. B-3 and B-4.) Precautions were taken to keep the road surface clean allowing a uniform rolling friction coefficient. Observations during tests showed the rolling resistance peaked at 600 - 1,100 lb during initial acceleration, then leveled off at 200 - 500 lb during the run.

The test setup used to evaluate self-level wind characteristics was also used to assess the property of abrasion resistance of the plastic braided rope. During early stages of testing it became evident the plastic braid had inferior abrasion resistance. To find means of improving abrasion resistance two cover materials were tried. These materials, in the liquid form, were applied to lengths of plastic braid and subjected to the same use profile as an untreated length. Coatings used were:

- o A polyurethane clear fabric impregnant
  Chemglaze Z003
  Hughson Chemical Co., Los Angeles, California
- o A neoprene rubber-like coating
  Hi-ten N-55 neoprene coating
  Gaco Western, San Mate , California

An additional parameter investigated at this time is the possibility of degraded breaking strength due to the coating applied. For this, select lengths of coated braid were retested for breaking strength.

RESULTS

Elongation

The elongation of each candidate rope is shown in the following tabulation:

Fig. B-3 Wind Up Test Set Up

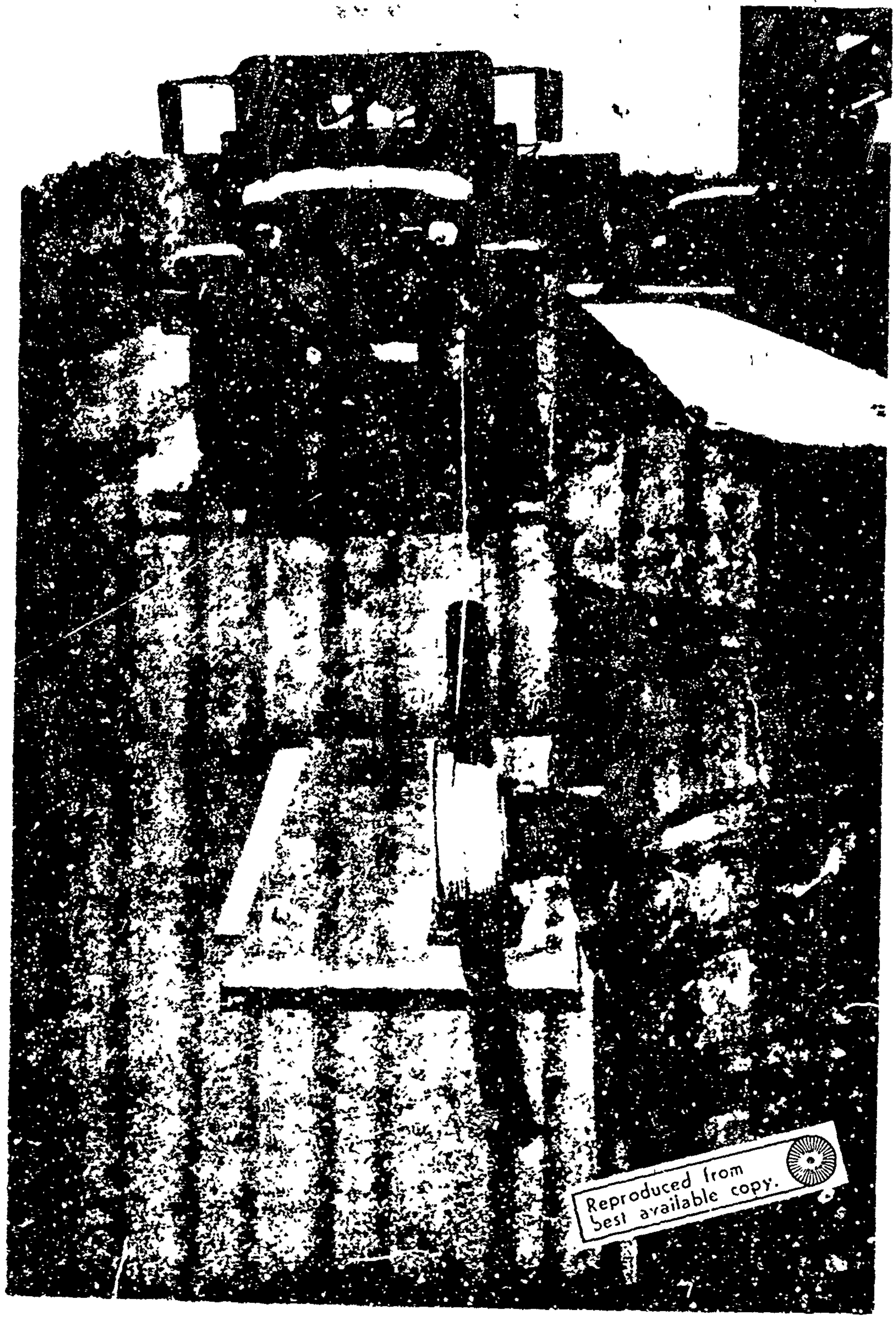


Fig. B-4 Wide Drum, Plastic Rope, 3.75° Fleet Angle

Material	Elongation $\frac{in.}{in.}$/lb (10^{-6})
Wire Rope (present)	6.25
Wire Rope (swaged)	5.12
Plastic Rope	114.2
Webbing	95.9

Permanent Deformation

An attempt was made to observe any possible permanent deformation of the materials tested. Each specimen was loaded to 1,000 lb and held there for approximately 30 min., at which time the load was released. This left a deformation in the plastic braid and webbing of 0.25 in. The wire rope had no measurable deformation. After 3 hours the candidate ropes were measured again and found to have returned to the original length.

Breaking Strength

Pull tests performed on each of the candidate ropes are tabulated below. Tests were run with the plastic braid with no coating and with polyurethane and neoprene coated samples.

Material	Breaking Strength, lb
Wire Rope - 3/16 diam.	3,700
Wire Rope Swaged - 0.208 diam.	Over 4,000
Plastic Braid - 5/16 diam.	
(a) Uncoated	2,500
(b) Polyurethane	1,500 - 2,300*
(c) Neoprene	1,800 - 2,200*
Webbing	Exceeded 4,000 (the capability of the test setup)

*These pull tests were performed on coated samples that had undergone ten cycles of self-level wind tests for abrasion resistance determinations.

The variation in breaking strengths was due to the rope failing at the entrance to the test fixture; with the snubbing type fixture used (Fig. B-1), as the rope is pulled in tension, it tightens on a drum and puts the rope fibers contacting the drum in compression. The uncoated rope is free to flow and move away from this compressive load while the coated rope is more resistant to this relieving capacity. Examination at the point of failure showed the compressive mode of failure.

An additional pull test performed on the plastic braid served to answer the question of "how many cut braids would induce failure?". For this determination several samples were pull-tested. The first had two cut braids; the second, four, and so on. Rope failure due to eight cut braids occurred at 2,200 lb.

It should be noted that all all but one test the failures occurred only to the outer braid coverings. The inner braided core held at about 1,000 lb after rupture of the braided cover. This includes the condition of reducing rope strength by cutting the braids.

Damping Characteristics

The property of damping is an important characteristic to have in a rope being used in a helicopter hoist. Its importance stems from the fact that rotating ropes tend to "blossom" or increase in their outer diameter. This can be a problem if fairleads are designed to only accept fixed diameter ropes. Also, in this helicopter hoisting application the downwash acting on the injured person being lifted can twirl the body aerodynamically. This can be resisted using a heavily damped rope.

The results of the torsional damping tests indicated the plastic braid rope had slightly better damping characteristics than the swaged wire cable and much better than the conventional wire cable. For example, the loaded samples were wound five times, then released, and the overtravel turns

noted; this was done in both directions. The plastic rope overtraveled 1-3/4 turns, the swaged wire cable overtraveled two turns, and the conventional wire cable overtraveled three turns; with a 10 turn windup, the plastic rope overtraveled 3-1/2 turns and the conventional wire rope overtraveled seven turns.

High Speed Capstan Tests

The results of the tests are shown in the table below:

Test Run	Speed (RPM)	Load (Lb)	Remarks
#1	5,250	50-100	Fairly even level wind
#2	8,000	75-150	Good level wind
#3	12,000	100	Slight bunching in center
#4	16,000	100-200	Poor level wind - bunching in center
#5	2,000	100-200	Same as #4
#6	25,200	100-200	Poor level wind - bunching up on edge

The results of the tests indicate the capstan type drive continues to function at the higher line speeds; however, the level wind on the storage drum was affected by the speed. The cable tended to bunch up in the middle or on one side at the higher line speeds; however, when the cable was wound off the storage drum, it came off in a satisfactory manner.

Self-Level Wind Capabilities

This characteristic was investigated to determine if proposed simplifications of present helicopter hoists could be realized. The hoist now in use incorporates a powered capstan for pulling up the load and a level wind mechanism to feed the wire rope into a storage drum. New design modifications are intended to do away with the capstan and level-wind mechanism.

The plastic rope, the swaged wire cable, and the conventional wire cable were tested to determine the level wind capability of each type. The tests were performed varying the sheave flange width, center distance to fairlead sheave, and reel-in speed. For the anticipated hoist design constraints, the plastic rope showed good level wind characteristics, the swaged wire cable had slightly less favorable characteristics, and the conventional wire cable has the worst properties. Table B-1 shows the results of the level wind tests.

The compliance of the plastic braid was a benefit to its level-wind ability in this particular test setup. Because of the manner used to apply load the compliant braid damped out load fluctuations. It was noted during testing the wire rope would tension up then go slack, thus producing an erratic level wind due to the pulsating winding tension. It is felt that the wire rope would exhibit a better tendency to level wind if it was pulling against a more constant load, that of gravity.

Abrasion Resistance

It was noted early in the test program that the candidate material would have to have good abrasion resistance properties. While performing the level-wind tests, note of the braid condition was made and, as suspected in this particular environment, the wire rope was superior to the plastic braid.

Two cover compounds were obtained that could be applied to the braid in an effort to improve its abrasion resistance. These were applied by hand and as a consequence were not uniform over the surface. Identical runs were made with coated and uncoated yacht braid. Coating is a definite aid to the abrasion resistance of the material.

Two coating materials were tried; one a clear polyurethane, the other a grey rubberized neoprene. Only the polyurethane caused a problem. That was, during unwinding the polycoated braid had a tendency to jam in the lay and

Table B-1

RESULTS OF SELF-LEVEL WIND TEST

Material	Run No.	Peak Load (lb)	Rope Speed (fpm)	Flange Width (in.)	Fairlead Center (ft)	Fleet Angle (deg)	Results
Stainless Steel Wire Rope 7 x 13 x 3/16 diam.	1	1,100	440	2	2	2-1/2	Poor, crowded one side (Fig. B-5)
	2	750	440	2	3	1-1/2	Fair, stuck on unwind
	3	800	620	2	4	1-1/4	Good (Fig. B-6)
	4	1,100	450	3	3	2-1/2	Fair, crowded one side
	5	800	500	3	4	1-3/4	Fair, crowded one side
Samson Brand 2 in 1 Stable	6	700	650	5	2	6	Good, used only 4 in. width
	7	650	450	5	3	4	Fair, did not fill full width
	8	700	800	5	4	3	Fair, crowded one side
	9	650	700	4	2	4-3/4	Fair, crowded one side
	10	600	750	4	3	3-1/4	Fair, crowded one side (Fig. B-7)
	11	750	700	4	4	2-1/2	Good, crowded one side
	12	775	700	3	2	3-1/2	Good, slight crowding
	13	650	650	3	3	2-1/2	Good (Fig. B-8)
	14	600	450	3	4	1-3/4	Good
Acco Swaged CRES Cable	15	900	450	2	2	2-1/2	Good, slight crowding
	16	1,050	600	2	3	1-1/2	Good
	17	1,000	600	2	4	1-1/4	Very good
	18	800	500	3	2	3-1/2	Poor
	19	900	600	3	3	2-1/2	Good, slight crowding (Figs. B-9 and B-10)
	20	900	500	3	4	1-3/4	Good

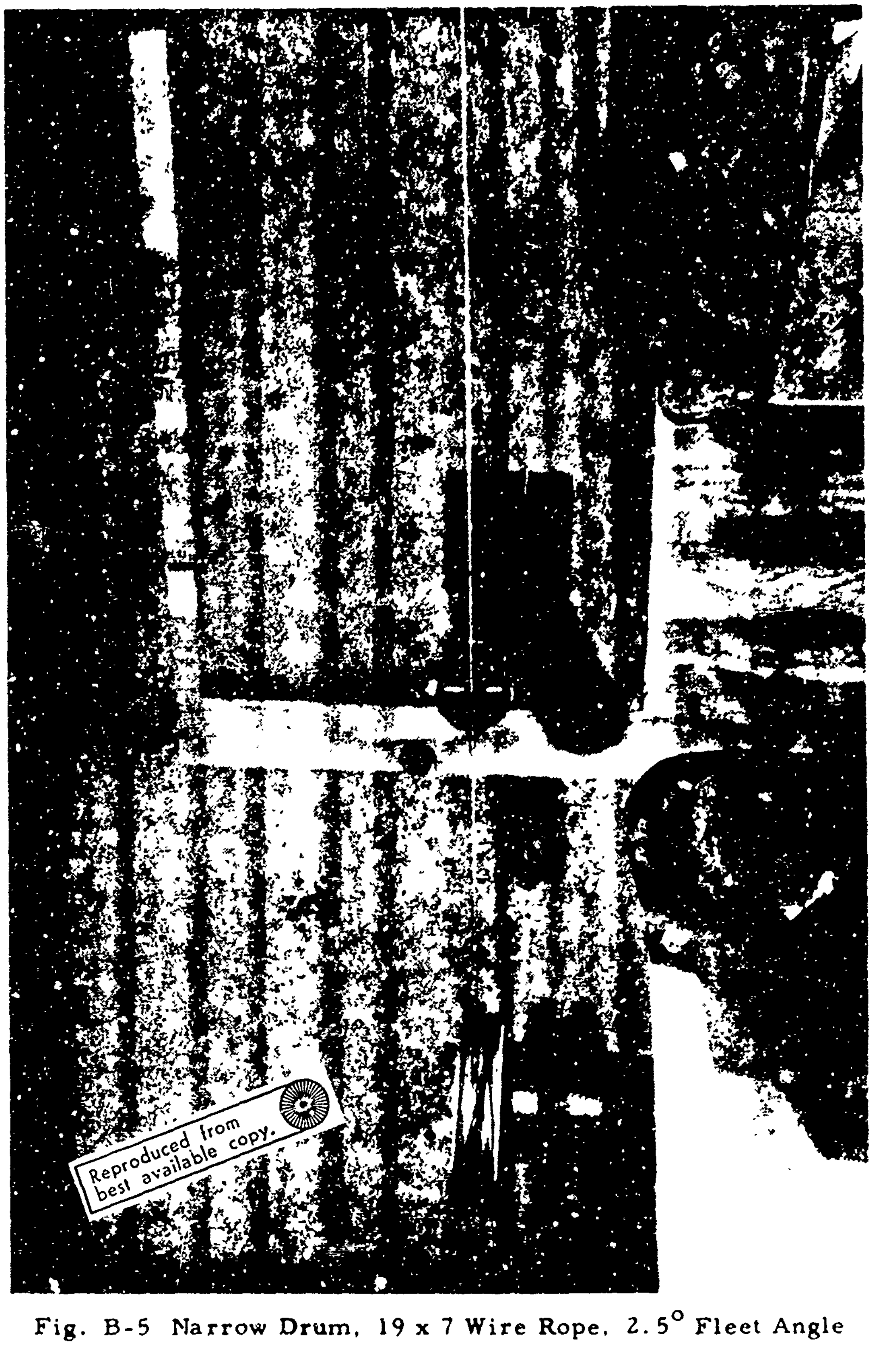


Fig. B-5 Narrow Drum, 19 x 7 Wire Rope, 2.5° Fleet Angle

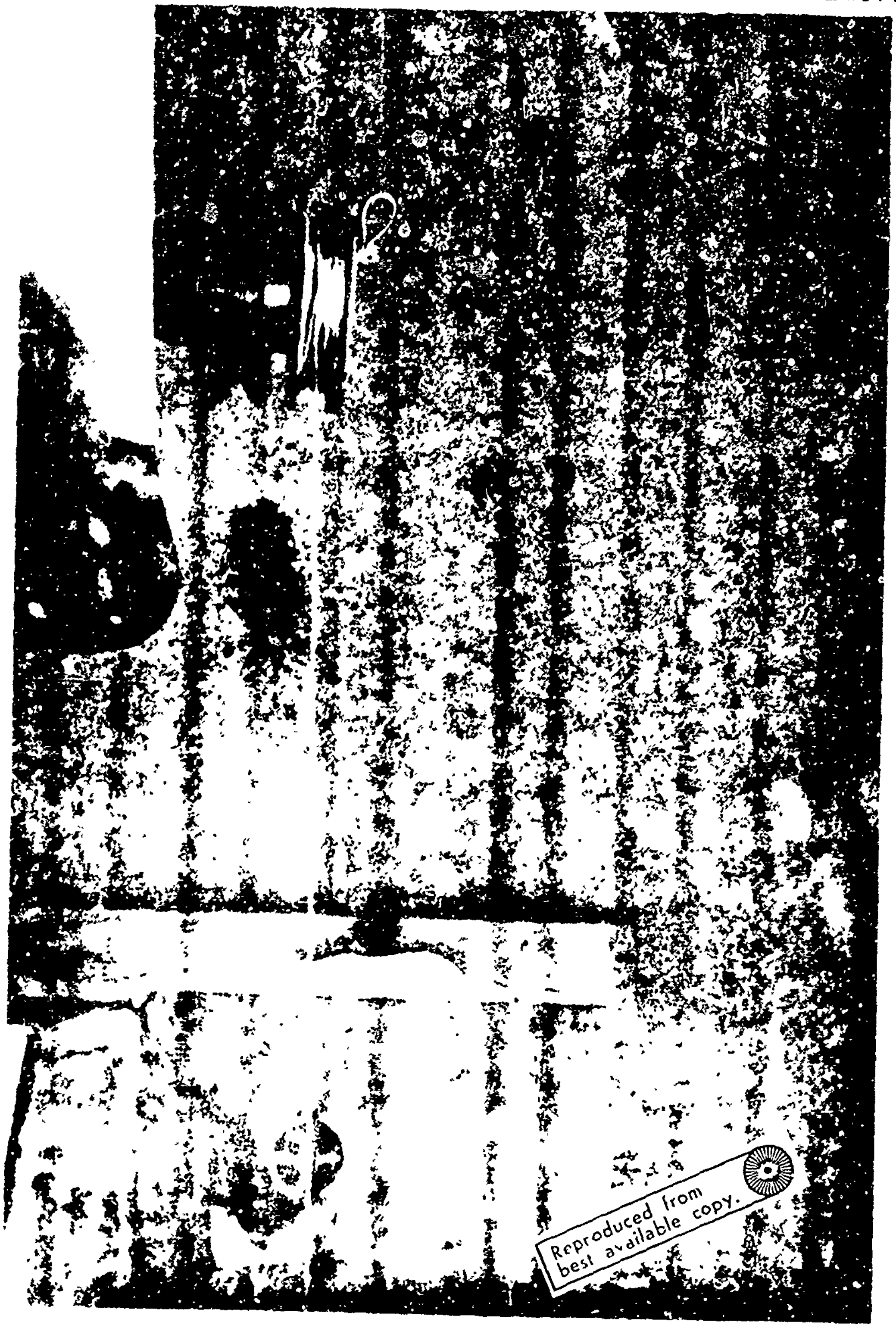


Fig. B-6 Narrow Drum, 19 x 7 Wire Rope, 1.25° Fleet Angle

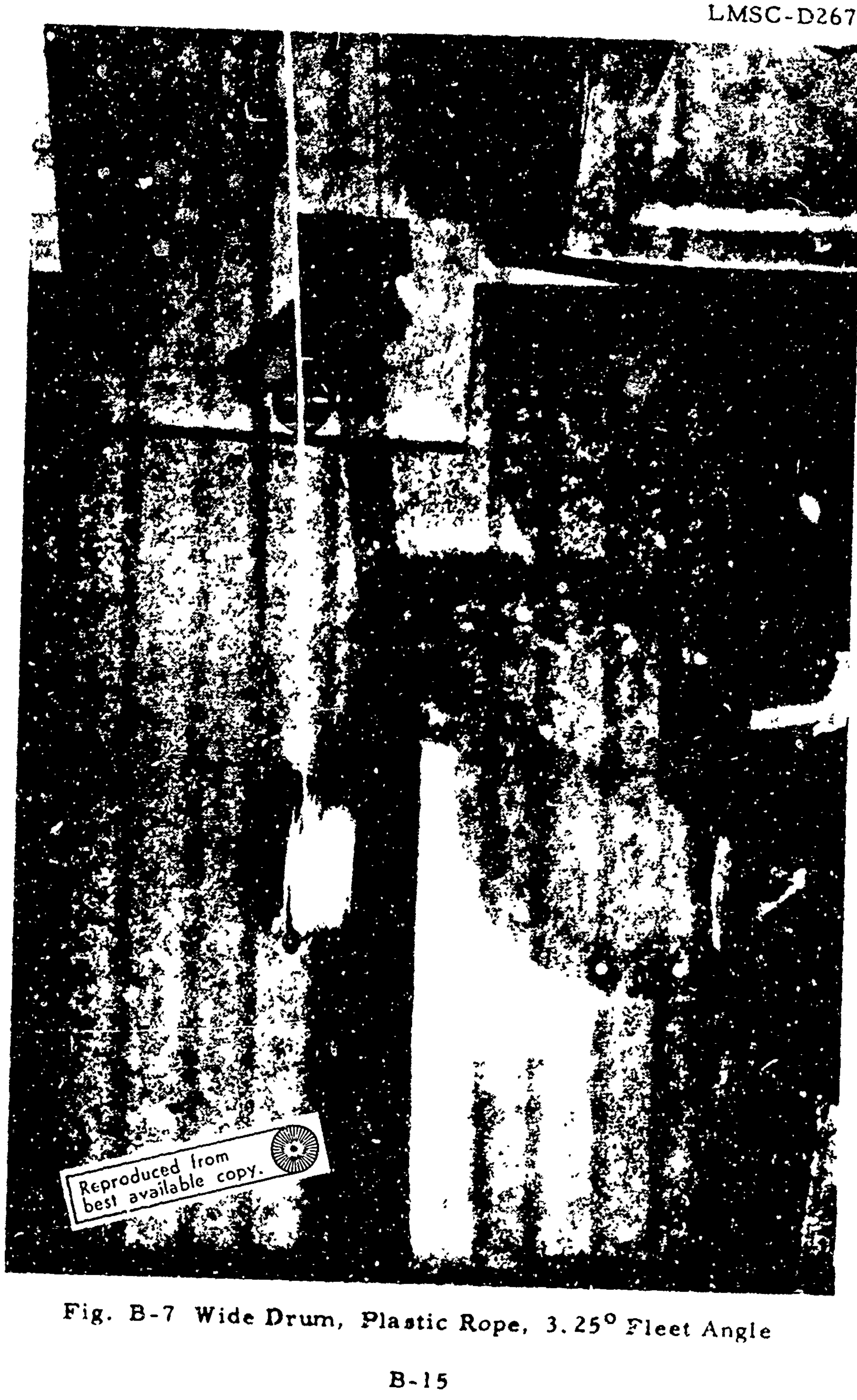


Fig. B-7 Wide Drum, Plastic Rope, 3.25° Fleet Angle

Fig. B-8 Medium Width Drum, Plastic Rope, 2.5° Fleet Angle

Fig. B-9 Medium Wide Drum, 19 x 7 Swaged Wire Rope, 2.5° Fleet Angle

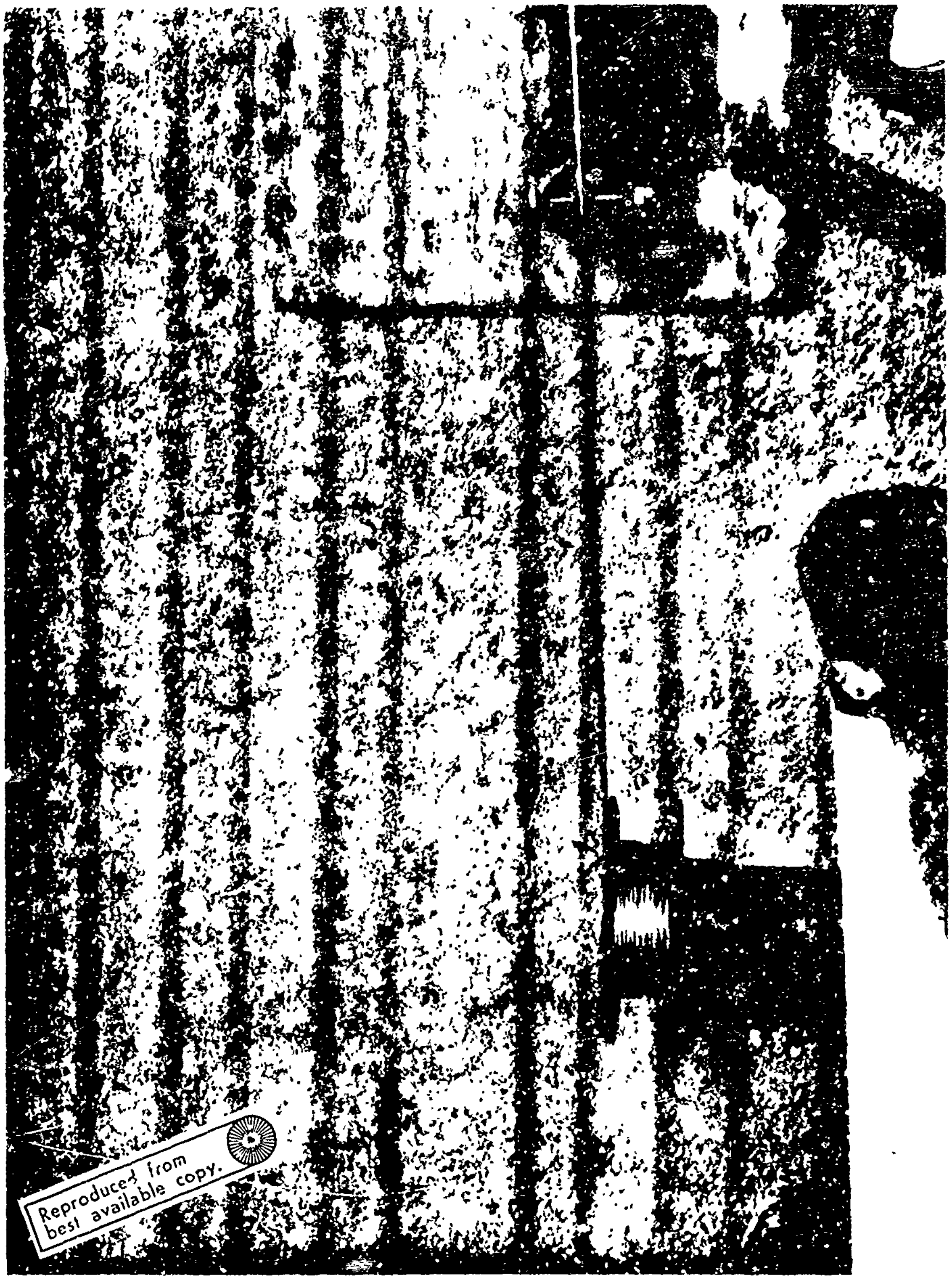


Fig. B-10 Medium Wide Drum, 19 x 7 Swaged Wire Rope, 2.5° Fleet Angle

began to reverse wind. Once, a force of less than 5 lb was required to unjam it; the second time, approximately 10 lb were necessary. This could present a problem in reeling out unless the hook load was more than 10 lb or the traction sheave kept the necessary tension on the line as it was reeled out. Results also indicate that a drum with a large radius at its flange base would resist "diving" of the rope as it winds on the drum. This jamming tendency was not observed with the rubberized neoprene coated braid.

Physical Properties

The maximum load to be hoisted will be 600 lb; therefore, the minimum breaking strength of the hoisting rope is 2,600 lb. This requires 5/16 in. diam. plastic rope which weighs 3 lb/ft, or 3/16 in. diam. wire cable which weighs 6.5 lb/ft. Thus, the plastic rope will take more volume.

The cost of the plastic braid at retail prices is 17 cents/ft on 300 ft lengths while the wire rope costs approximately 34 cents/ft.

Field reports have indicated that the failure of the guillotine to operate when required has created serious problems during missions. The plastic rope has an advantage over the wire cable in that the rope could be readily severed with a knife while the wire cable could not.

CONCLUSIONS

results of the tests indicate that both the plastic braided rope and the wire cable have desirable properties which make them candidates ce the present 19 x 7 CRES non-rotating wire cable.

The self-level wind characteristics of both candidates were better than those of the present cable; however, the plastic rope showed better characteristics than the swaged cable at the larger fleet angles.

The damping characteristics of the plastic rope and swaged wire cable were very similar; both were better than those of the present wire cable.

The elongation of the plastic rope was 7 percent; the wire cable elongation was less than 1 percent.

The abrasion resistance of both of the wire cables is much better than that of the plastic rope. The uncoated rope had poor abrasion resistance; however, this property was improved with the use of a polyurethane coating. Additional testing should be done on the plastic braided rope with the coating commercial impregnation methods, which would insure more uniformity of penetration and outside coating thickness.

Appendix C

MILITARY CHARACTERISTICS FOR HIGH SPEED UTILITY HELICOPTER HOIST

1. REQUIREMENTS:

a. Provide U. S. Army medical evacuation helicopters with a high speed method of lifting a patient to the helicopter from areas where a landing cannot be made.

b. Source of Requirement:

(1) USARV Letter, CDCCS-LV, 13 Dec 70, subject: DPSDR for a Triple Canopy Resupply System (TRICARS).

(2) FONECON, MAJ Cloke, CDC Medical Service Agency, 4 Aug 71.

(3) COL Shane, CO Aero Medical Laboratory, Visit to USALWL, 5 Aug 71.

2. OPERATIONAL AND ORGANIZATIONAL CONCEPTS:

a. Operational Concept: Aero medical evacuation units would use this device to rapidly lift wounded and injured personnel from the ground to a helicopter in areas where the helicopter cannot land.

b. Organizational Concept: It is envisioned that item would be available to using units through normal supply channels for the class of supply and issue on a one-for-one replacement basis for the current hoist on the same basis of issue.

3. JUSTIFICATION AND PRIORITY:

a. Reason for the Requirement: The UH-1 Helicopter International Rescue Hoist currently used to lift personnel to the helicopter in areas where it cannot land operates very slowly and requires the helicopter to hover for excessive time at an altitude where it is extremely vulnerable to enemy ground fire and where engine failure could be disastrous.

b. Priority for Requirement: This problem is not included in the May 1971 MACV Significant Problem Areas Report but it meets the criteria for Priority Group II.

4. CHARACTERISTICS:

a. Physical Characteristics:

(1) Maximum Num' er of Major Components in this Unit: (Essential) 4 (Flywheel, motor, drum with cable, and hoist arm assembly)

(2) Maximum Weight: (Essential) 180 lb installed

(3) Cubic Measurements: (Essential) Occupy no more usable space than that occupied by the present hoist.

(4) Installation: (Essential) Be capable of being installed on existing mounting points at the same level of maintenance in the same amour of time as the present hoist.

(5) Maximum Hoist Power Requirement: (Essenti.l) The electrical energy must be supplied from aircraft engine accessory generator system and must not exceed the power requirements for the current hoist.

(6) Transportability: (Essential) Air

(7) Expendable: (Essential) No

(8) Environmental Requirements: (Essential) Categories 1-8, AF 70-38.

(9) Compatibility: (Essential) Be capable of being operated on the UH-1D and UH-1H helicopters; (Desired) the Utility Tactical Transport Aircraft System (UTTAS).

(10) Storage (Shelf Life): (Essential) 5 years.

b. Performance Characteristics: (Essential) Have the following improved characteristics without degrading any of the performance characteristics of the existing hoist:

(1) Lifting and Lowering Sustained Operation Rate: (Essential) Be capable of operating at all speeds up to its maximum and performing at the following worst case scenarios:

Scenario	No. of Hoist Operations				Wt Hoisted (Lb)
	200 Lb	400 Lb	600 Lb	Total	
A	9	0	0	9	1,800
B	7	1	0	8	1,800
C	6	0	1	7	1,800
D	5	2	0	7	1,800

(2) Mission Scenario Time: (Essential) Be capable of performing the four worst scenarios within the following times (based on the assumptions listed:

Scenario	Maximum Time
A	12 Minutes
B	11 Minutes
C	10 Minutes
D	10 Minutes

Assumptions:

A. Dwell Times at Top and Bottom are the following:

Load	Dwell Time
200 lb	0.25 min.
400 lb	0.50
600 lb	1.00

B. No Dwell Time is required at top prior to the first or after the last hoist.

(3) Acceleration/Deceleration: (Essential) Exert not more than a 1 "g" force (32.2 ft/sec on the load being lifted or lowered).

(4) Mission Reliability: (Essential) A mission reliability of 50 percent is required; (Desired) 95 percent. System must have a 0.99999 reliability against catastrophic failure during lowering and retrieving of personnel.

c. Maintenance Concept:

(1) (Essential) Only standard Army POL products must be required for lubrication; (Desired) a lifetime lubrication system should be used.

(2) (Essential) The aircraft crewchief must be capable of determining the state of operational readiness by simple visual inspections and

operational tests. Cable check/preventative maintenance shall be accomplished by oiling and cleaning with a rag and looking for broken strands.

(3) (Essential) Crewchief maintenance must be simple and consist of component part replacement and minor repairs without using special tools.

(4) (Essential) The total maintenance effort must not exceed one hour of maintenance for each 400 cycles of hoist operation.

d. Human Engineering/Safety Characteristics:

(1) (Essential) Be safe in operation in accordance with AR 385-16, dated 11 Feb 67.

(2) (Essential) A visual means will be provided to indicate how much cable is extended.

(3) (Essential) The hoist must automatically slow down and stop when the cable is fully extended or retracted.

(4) (Essential) When the cable is retracted, the hoist must not change any of the flight characteristics of the UH-1D/H helicopter on which it is installed. The aircraft must be capable of limited vertical and horizontal movement with the cable extended.

(5) (Essential) The system will include provisions whereby the extended cable can be rapidly cut at the helicopter without backlash and/or unwinding of the cable damaging the aircraft or endangering the crew.

(6) (Essential) When in operation, the loaded system will remain within the fore, aft, and lateral CG limits of the aircraft on which mounted.

(7) (Desired) A means to lower and raise loads manually will be provided.

e. Priority of Characteristics:

(1) Safety (Not change flight characteristics or CG).

(2) Safety (Cable cutting capability).

(3) Performance (Mission reliability).

(4) Performance (Speed).

(5) Performance (Sustained Operation).

(6) Performance (Acceleration/Deceleration).

5. <u>PERSONNEL CONSIDERATIONS</u>: Introduction of this item into the Army inventory will require no additional personnel spaces in TO&E of tactical units.

DATE ____________________ APPROVED ________________________